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DAVIDSON LABORATORY

Report 1382

STUDIES OF OFF-ROAD VEHICLES IN THE RIVERINE ENVIRONMENT
Vol. I, Performance Afloat

by

I. R. Ehrlich
I. O. Kamm
and
G. Worden

Prepared for the
U. S. Army Tank-Automotive Center
under
Contracts DA 30-069-AMC-789(T),
DAAE-07-69-C-0356,
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Approved

I. Robert Ehrlich, Manager
Transportation Research Group

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ABSTRACT

As the first of the three volumes of the publication entitled "Studies of Off-Road Vehicles in a Riverine Environment," this report attempts to assemble the vehicle-environment relationships which may be employed in the design of swimming vehicles. It is intended to be used as a guide for engineers who are basically automotive designers. Therefore, many generalizations and approximations have been developed which would not be acceptable to naval architects, but which should be extremely useful to an automotive engineer.

KEYWORDS

Amphibians
Swimmers
Floaters
Off-road vehicles
Propulsion
Drag
Riverine environment
Mobility

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(Cont'd)

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FOREWORD TO VOLUMES I, II, AND III
of
STUDIES OF OFF-ROAD VEHICLES IN THE RIVERINE ENVIRONMENT

One of the most important considerations in the development of military vehicles has been the recent recognition that to achieve true cross-country mobility a vehicle must possess the inherent capability of crossing inland waterways. This fact has brought a new dimension to the problems confronted by those faced with the responsibility for the design and development of new military vehicle concepts. In order to incorporate this new facet into future designs, it is quite logical to turn for guidance to the designers and developers of amphibious vehicles, who have for some time been confronted by similar problems. It is in this context that the studies discussed in these three volumes were undertaken. Although the purpose of the work was originally intended to aid in the design of $\frac{1}{4}$-ton floating vehicles,[1] the information presented is applicable to land vehicles in general; hence, this document may be considered as a guide to methods for evaluating all types of land vehicles relative to their performance in crossing inland waterways.

There are two distinct aspects of vehicle evaluation:

(1) First, it is necessary to determine functional relationships between the vehicle and its environment so that its performance capabilities may be expressed in quantitative form.

(2) Second, to determine the vehicle's actual effectiveness, it is then necessary to compare these performance capabilities with appropriate quantitative environmental attributes of the inland waterway population (or some sub-set thereof of military interest). In view of the present state of knowledge in vehicle mechanics and environmental sciences, the attainment of these objectives represents a tremendous task. The work presented in this report constitutes a compendium of much of applicable knowledge in this area, with progress naturally more advanced in some areas than others, and it provides, moreover, a meaningful conceptual framework for continued work by indicating the knowledge gaps that exist.

Three different modes of locomotion are important in connection with the general stream-crossing maneuver: fully floating, fully land-borne, and water-land transition. Of equal importance are the associated environmental factors relevant to all three modes.

At its inception, this study was designed to concentrate most heavily on various aspects of the fully floating mode,[2] and much of the effort was directed toward the development of vehicle-environment relationships for the fully floating case. Major consideration has been given to such factors as hydrodynamic resistance, propulsion effectiveness, freeboard requirements, and the effects of waves. These are areas in which a sophisticated technology has already been developed by the naval architect,[3,4] but where applications to the design and evaluation of amphibious vehicles or ships having vehicle-like forms have been extremely limited. As the work progressed, however, it became increasingly clear that by far the most critical element of the stream-crossing maneuver was egress. The study of egress is therefore of prime practical importance.

This water-land transition aspect of stream-crossing is a subject about which very little is known. Analysis of the "twilight zone" is extremely complicated, involving consideration of both hydrodynamical and terramechanical factors. Many actual field problems, however, have demonstrated that the transition phase, particularly egress, is usually the most difficult element of the entire stream-crossing maneuver; hence this complication cannot be avoided.

Relationships between the fully land-borne vehicle and its environment (terrain) have been under intensive investigation for many years,[5,6,7,8] notably under the auspices of the U. S. Army. The task of particularizing the previously developed models to the bank environment where the proximity of the stream or river is known to affect the soil and vegetation is beyond the scope of this study.

Another area in which the current efforts represent only a scratch on the surface of the problem is that associated with the determination of environmental attributes. It has become apparent that complete quantification of world-wide stream properties in sufficient detail for vehicle

performance evaluation cannot possibly be achieved by direct measurement within the scope of any reasonable study, nor is the military really interested in the characteristics of every stream in the world. Therefore, an approach toward categorizing waterways on the basis of available climatic, geological, and other existing environmental data has been initiated as part of the study, so that those of military interest may be studied.

It is worth repeating here that the present work is by no means regarded as a finished product. Many of the relationships which are presented should be considered as no more than reasonable hypotheses which must be validated and, if necessary, modified on the basis of carefully controlled experiments. Much additional research in the various problem areas and a major effort to coordinate the various facets into a comprehensive systems analysis is required before the kind of evaluation-scheme envisioned as the long-term goal of this study can be achieved.

This document (Vol. I) represents a part of the entire multi-part study to which this foreword is an over-all introduction. Volume II is concerned with water-land transition, and Volume III with associated environmental factors.

It should be noted that Hydronautics, Inc. is preparing an amphibious-vehicle design manual for the U. S. Army Materiel Command, which should be published at about the same time as this document. Coordination has been established between Stevens and Hydronautics, and in general the two documents should not overlap, but rather complement one another.

INTRODUCTION TO VOLUME I

The aspect of vehicle-environment relationships on which most effort has been expended in the past, and of which most is therefore known, is that dealing purely with fluid dynamics. Actually, a fully floating body is immersed not in a single fluid but in two; hence there are initial complications. This study will neglect the relationships between the vehicle and air, because air is relatively insignificant at the general range of speeds considered.* The complete analysis of the very fast vehicles (hydrofoils and captured-air vehicles) is considered to be beyond the scope of the present work.

This volume attempts to assemble those established vehicle-environment relationships which have already been developed in other areas of research and which are applicable to swimming-vehicle design. To make the report as useful as possible to the design engineer, many generalizations and approximations have been employed (enough perhaps to astonish the "purist"). These are of practical interest to the amphibious-vehicle designer, and provide guidance.

The authors wish to acknowledge the contribution of Mrs. Ann Ljone, who made a significant contribution by reviewing the report to ensure the accuracy of sections concerned with naval architecture.

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*Wind loads which affect safety are considered, however.

NOMENCLATURE FOR VOLUME I

A	area
A_F	static submerged frontal area
A_{SW}	submerged wetted area
B_o	center of buoyancy
b	width
C_B	block coefficient
C_D	drag coefficient
C_K	Kempf roughness coefficient
C_f	coefficient of skin friction
D	total drag
D_f	drag force due to skin friction
d	vehicle draft
F	force
F_B	buoyant force
Fr_L	Froude number, based on over-all length

$Fr_{\sqrt{A}}$	Froude number based on the square root of the static submerged frontal area
f	freeboard height
G	center of gravity
g	gravitational constant
h	wave height (double amplitude)
I	moment of inertia
K	radius of gyration
K_d	damping coefficient
L	the over-all front (bow) to rear (stern) vehicle (boat) length
ℓ	length
ℓ	a characteristic length
M	metacenter
$\overline{MG}$	metacentric height
N_{Fr}	Froude number
Q	water flow rate
r	transverse metacentric radius
T	moment or torque

t	period
V	volume
v	velocity
W	weight
x	distance
β	righting moment
γ	specific weight
Δ	displaced weight of water
δ	rudder angle
η	efficiency
θ	roll angle
λ	wavelength
ν	fluid kinematic viscosity
ρ	fluid density
τ	time
φ	wave slope

Chapter 1
FLOATABILITY

Any vehicle design is a grouping of many compromises. Most land vehicles are relatively short and wide, with a length/width ratio of about 1.5 to 2.0. This ratio has been developed in a reasonable compromise between vehicle control and vehicle ability to negotiate obstacles. Vehicle height is limited for side-slope stability, but good ground clearance dictates a high body floor. Wheels and/or tracks are located to give good brake angles and angles of approach and departure; their size and construction has to be such as to ensure good mobility, but they cannot be so large as to make the vehicle unwieldy or, conversely, to use space at the expense of useful load. Any shape which becomes excessively long must be divided into several sections, as in articulated or trained (coupled in tandem) vehicles.

Compromises are no less present in amphibious-vehicle design, for all the conflicting problems inherent in the design of land vehicles are present, plus additional problems arising from the requirement for floatability and adequate self-propulsion in water. Most land vehicles are too dense to float.* Special measures, therefore, must be taken to decrease their density (weight/volume) or increase their water displacement.

To provide flotation, hulls or external displacement bodies are usually added to a basic land vehicle. More often than not, these flotation devices (even integral hulls) are impediments to land travel. To reduce the impedition, some vehicles are provided with detachable flotation devices. These, however, are usually bulky and therefore awkward to stow on the vehicle. The user is then faced with the dilemma of carrying the flotation kit for the life of the vehicle, so that it will be available for the relatively small amount of time that it will be called into use,

*Many land vehicles will float if the total peripheral area is sealed against water entry.

or of transporting the devices separately and risking their not being available when needed.

If the amphibious craft is to have sufficient roll stability in water, it must be wider than its purely land-borne counterpart. If it is to float during water entry and exit, at adequate angles, it should be longer or higher. In any case, land mobility and maneuverability suffer.

Amphibious vehicles may be divided roughly into two groups: floaters and amphibians. The line of demarcation between the two is rather arbitrary. However, the term "floater" is usually applied to basic land vehicles which have minimum floating ability and can propel themselves but slowly or not at all through water. Amphibians are usually thought of as vehicles which have been designed for operation both on land and in water; hence, in water, they perform much better than floaters (and on land quite often much more poorly). They achieve higher water speeds and can operate in waves and surf.

The floater fulfills the most basic needs for water operation. Its flotation is usually provided by a hull, a plastic form, air bags, or any other auxiliary device which displaces a volume of water in some excess of its weight. The vehicle is usually propelled by its own wheels or tracks. Little attempt is made to provide it with more than a basic ability to move. The top speed of such a vehicle is usually in the neighborhood of 2-3 mph. Typical examples are the XM437 GOER, the XM410 ($2\frac{1}{2}$-ton 8 x 8), the M151 ($\frac{1}{4}$-ton 4 x 4) with flotation kit, the M656 (5-ton 8 x 8), and the M561 (Gamma Goat).

In general, floaters are rather unsatisfactory water vehicles. They perform satisfactorily only in the most calm and sheltered of inland waterways, and are useful only as a military expedient.

An amphibian is designed to be capable of operation in all waters, including open seas, surfs, and swift-running rivers. It usually offers protection from the elements for both crew and cargo. Auxiliary propulsion devices are generally provided, so that higher water speeds can be obtained. Usually some of the amphibian's capabilities for operating on land are sacrificed to achieve improved water performance; hence it is much more economical for amphibians to operate only in and near the beach or on

hard-surface roads.

The characteristics of amphibians are quite varied. The vehicles can range from the most simple displacement hull with wheels affixed to the highly sophisticated hydrofoil craft with retracting foils and wheels. Newest entry into the amphibian class is the surface-effect (air cushion) vehicle.

Although it is essential that all floating vehicles be able to displace their weight in water for static flotation, many employ the dynamic characteristics of water to obtain additional vertical forces which reduce the submerged volume and the correspondent vehicle drag. The two most common are the planing hull and the hydrofoil craft.

BUOYANCY

Archimedes discovered that the buoyant force acting on a body floating or submerged in water is equal to the weight of the volume of the water displaced.

The buoyant force is always directed vertically upward.

$$F_B = \gamma V \tag{1}$$

where F_B = buoyant force

γ = specific weight of water

V = displaced volume of water

Frequently, the displacement (Δ) is used when discussing boats.

$$\Delta = \gamma V \tag{2}$$

If all sides of the vehicle are rectilinear, then to determine the displacement of the submerged portion it is sufficient to multiply the length, width, and depth. But when the submerged surfaces are tapered,

the water displacement is less than that of a right parallelepiped with the vehicle's dimensions. For an estimation of the degree of "fullness" of the submerged portion without projections, a "block" coefficient C_B is introduced, showing what part of the parallelepiped represents the water displacement of the vehicle.

The displacement volume is then given by

$$V = C_B Lbd \tag{3}$$

where C_B = block coefficient

L = vehicle length

b = vehicle width

d = draft

The value of C_B depends on the type of vehicle, but may be estimated equal to 0.7 - 0.85 for amphibious vehicles of conventional design (see Fig. 1). Vehicles with especially large wheels, such as the Airoll, GOER, etc., should not use this figure.

The length L is measured at the loaded waterline, and should be distinguished from the over-all length which is measured between extreme points of the body. The width b is also measured at the loaded waterline. The draft, d, is the depth of immersion below the waterline of the hull, without consideration of any projecting parts (tracks, wheels, propellers, rudders, etc.). It is measured amidship of the vehicle, by taking a vertical distance from the deepest surface to the loaded waterline. The total draft is the maximum submergence of the vehicle with all projecting parts taken into account.

The ratios L/b and b/d of the basic hull dimensions suggest the vehicle's shape and give qualitative indices for this shape. For example, the ratio L/b affects resistance to movement; resistance decreases with an increasing ratio. The ratio b/d characterizes the vehicle's stability, and influences the magnitude of water resistance; with an increase in this

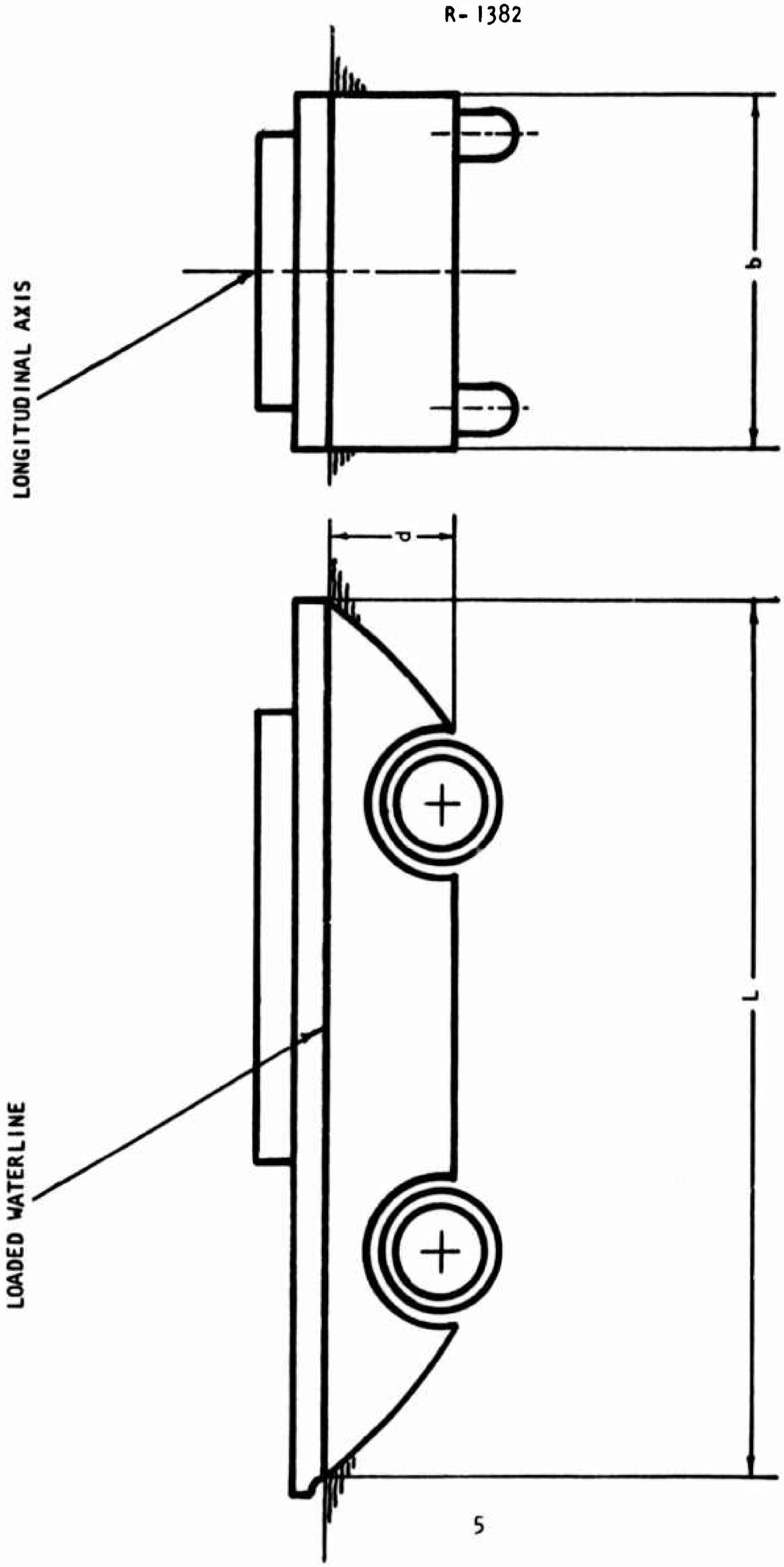


FIGURE 1. Basic Dimensions of an Amphibious Vehicle

ratio, stability increases (see following sections).

The load capacity, or deadweight, of the vehicle is its ability to carry a certain amount of load and be submerged by it up to the loaded waterline. Load capacity coincides with the difference between the entire water displacement of the loaded vehicle versus that of the empty vehicle.

Center of Buoyancy

The center of buoyancy, or the point where the buoyant force acts on a body, is located at the centroid of the displaced volume.

$$\bar{x} = \frac{1}{V} \int x dV \tag{4}$$

where $\bar{x}$ = distance from any arbitrary reference axis to the centroid of displaced volume of water

x = distance from arbitrary reference axis to dV

dV = differential displaced volume

In principle, Eqs. (1) and (4) are quite simple. However, their application to floating bodies of complex shape (such as amphibious vehicles) is, though not difficult, quite tedious. The most practical approach is to divide the vehicle into many simple shapes whose properties are well known or easily determined.

Consider a simplified vehicle such as is shown in Fig. 2. The displaced volume derived from the cargo body is $a \cdot b \cdot d$; that derived from the engine compartment is $d \cdot b \cdot c$; and that derived from each of the four wheels is $\frac{\pi f e^2}{4}$. Therefore, from Eqs. (1) and (3), we get

$$F_B = C_B \gamma V \approx 0.775 \cdot 62.4 \, (abd + cbd + \pi f e^2) \text{ lb} \tag{5}$$

Frequently, the problem is to determine at what depth the vehicle will float for a given weight. In this case, F_B is equal to the weight,

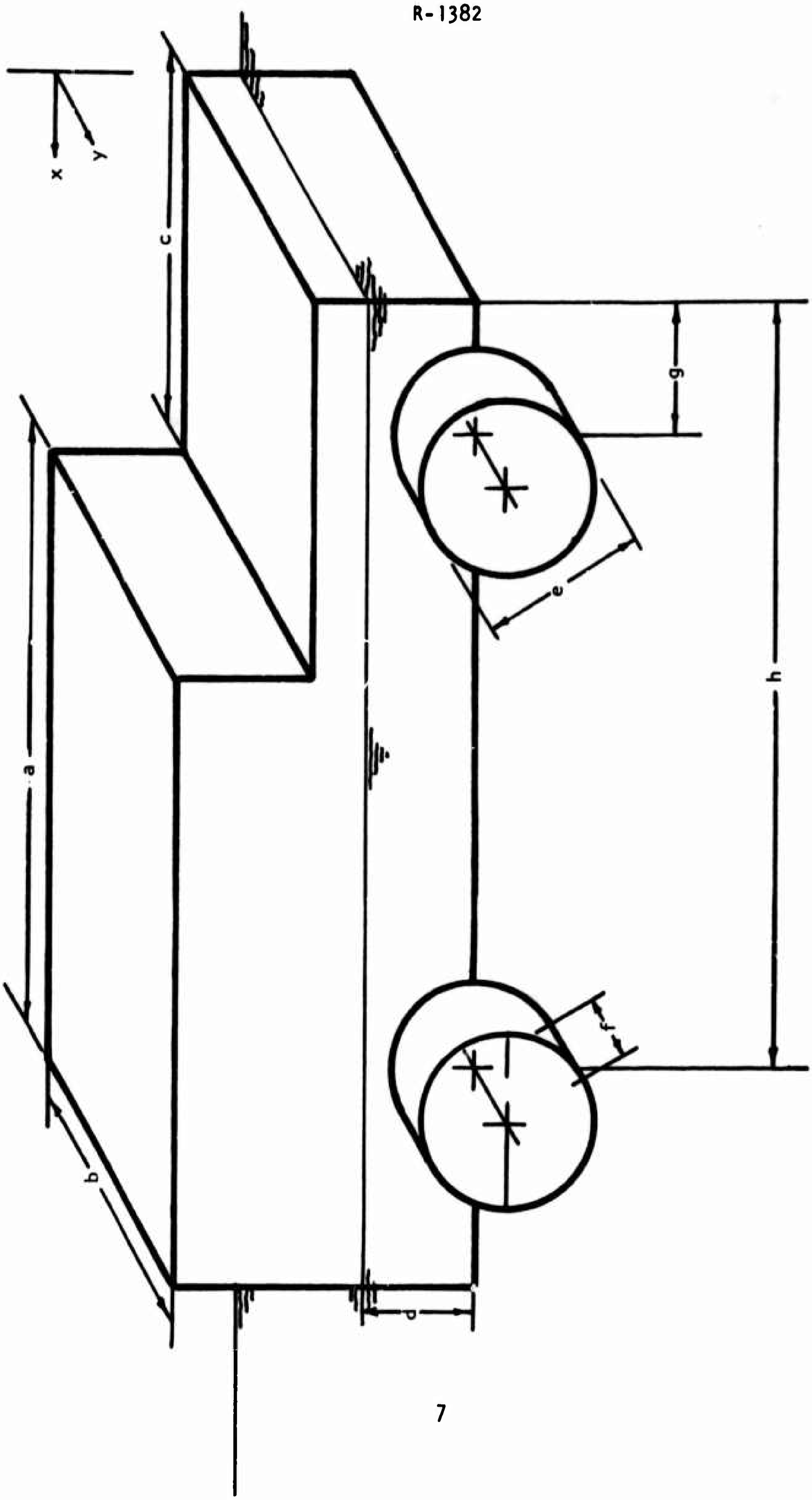


FIGURE 2. Dimensions Used to Estimate Buoyancy

and the problem is to solve Eq. (5) for the value of d .

To compute the point of action of this buoyant force, we take moments about the forward edge and rewrite Eq. (4):

$$\bar{x} = \frac{1}{V} \sum x_i V_i \approx \frac{\frac{c}{2} (dbc) + (\frac{a}{2} + c)\ (abd) + g(\frac{\pi f e^2}{2}) + h(\frac{\pi f e^2}{2})}{abd + dbc + \pi f e^2} \tag{6}$$

In this case, $\bar{x}$ is the distance of the center of buoyancy from the forward edge of the vehicle. Symmetry usually determines the distance of the center of buoyancy from the side of the vehicle ($\bar{y}$), as b/2.

Righting Moments and Stability

In the sense that it is considered in this section, stability is that condition which holds when a vessel floats upright and, if displaced at a small angle, returns to the upright position. The moment tending to return the vessel from any displaced angle is called the righting moment.

When the center of gravity of any floating object is below its center of buoyancy, the object floats in stable equilibrium. Certain floating objects, however, are also stable when their center of gravity is above the center of buoyancy; the stability and righting moments for such bodies may be determined by using the equations developed below.

Figure 3a is a cross section of a simple floating cube. The center of buoyancy, B_o , is at the centroid of the displaced volume of water (which in this case is at the centroid of the rectangular area ABCD below the surface). Let the center of gravity, G , be above the center of buoyancy. When the body is tipped, as in Fig. 3b, the center of buoyancy shifts to the centroid B_o' of the trapezoid A'BCD'. The buoyant force acts upward through B_o'. The weight of the body acts downward through G , which does not change position.

The point at which the vertical line through B_o' intersects the extension of the line B_oG is called the metacenter, M . For small angles

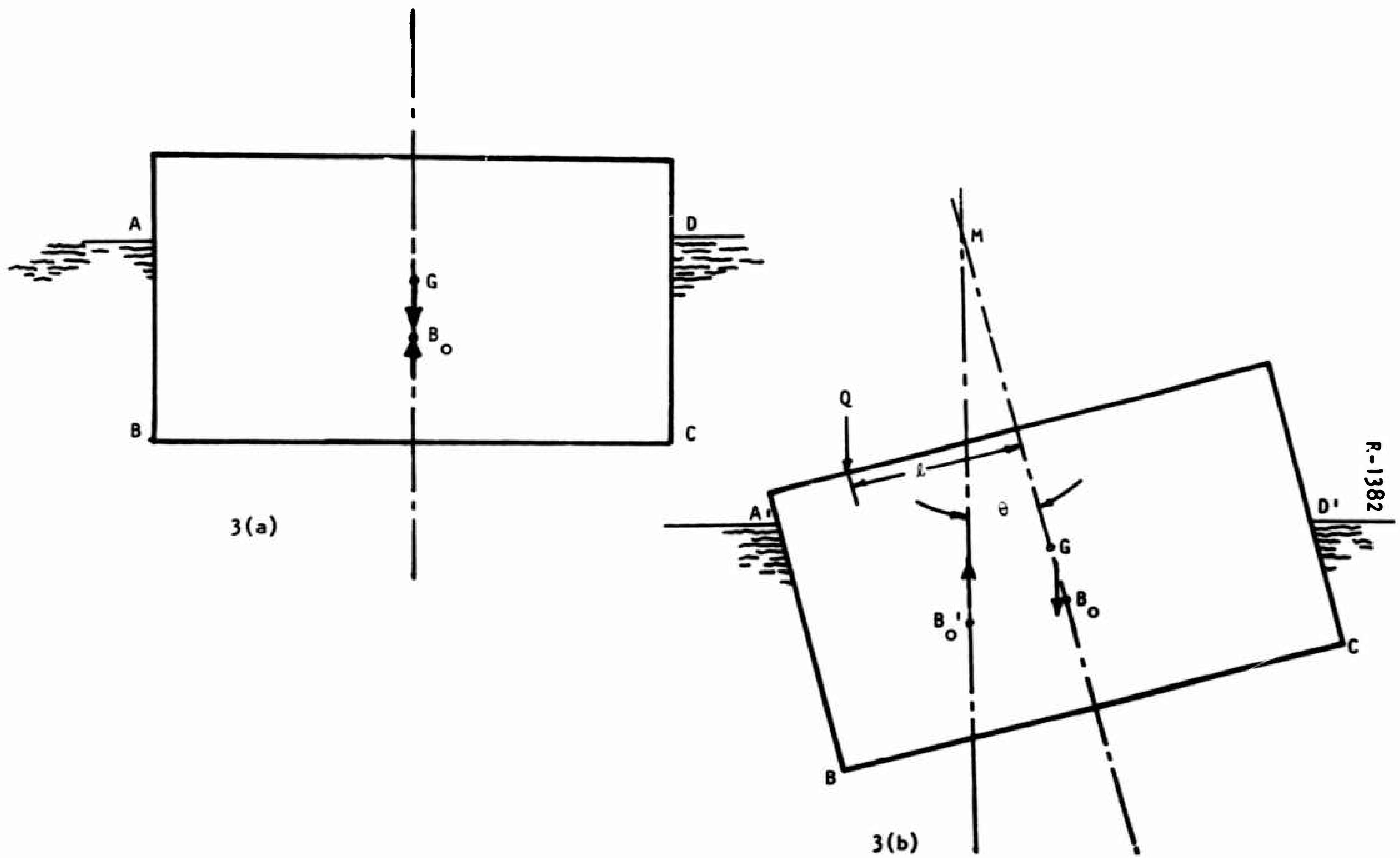


FIGURE 3. Stability of a Floating Object

of roll, the location of M remains constant. If M is above the center of gravity, G , the couple generated by the forces at G and B_o tend to return the body to the upright position, and the body is stable. When M is below G , the body is unstable. When M and G coincide, the body is neutrally stable. The distance $\overline{GM}$ is called the metacentric height. Since $\overline{GM}$ is relatively constant for small angles of roll, for most reasonable shapes, it is a direct measure of the stability of the body.

The restoring couple, or righting moment, may be calculated from

$$\beta = W \, \overline{GM} \sin \theta \tag{7}$$

where β = righting moment

W = weight of the body (or Δ; see Eq. 2)

θ = angular roll displacement

$\overline{GM}$ = metacentric height

Equation (7) applies to any angular displacement, in the determination of righting moments, but the distance $\overline{GM}$ usually changes with large roll angles. The value of metacentric height usually quoted for a vehicle (see table on page 14) is that value for an angular displacement θ near zero.

Another equation for determining metacentric height, on the basis of water-plane area, is

$$\overline{GM} = \frac{I}{V} \mp \overline{GB} \tag{8}$$

where $\overline{GM}$ = metacentric height

$\overline{GB}$ = distance between center of buoyancy and center of gravity

V = displaced volume of water

I = moment of inertia of the water-plane area about the fore and aft axis of the body $= \int_A x^2 \, dA$ (see Fig. 4)

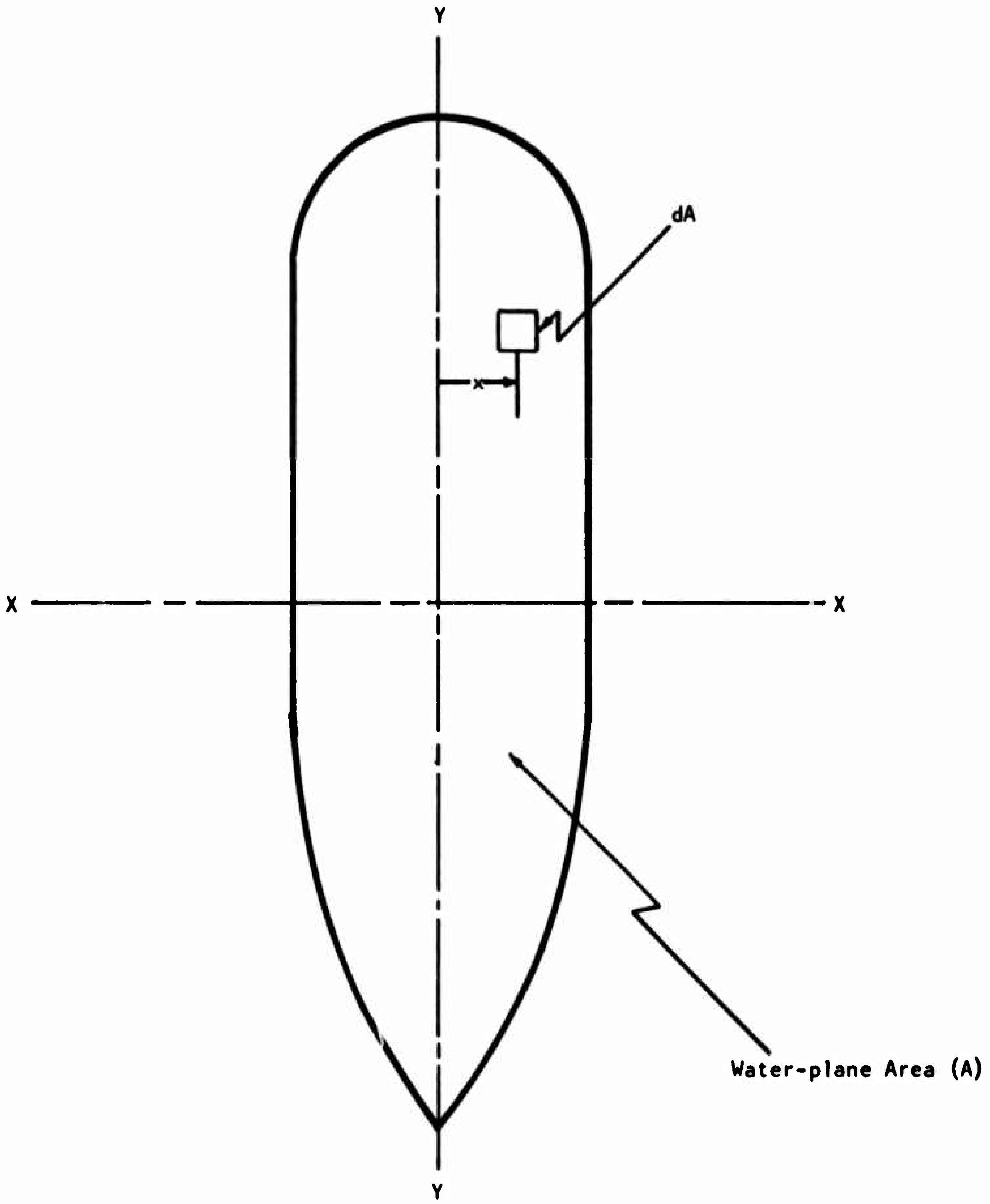


FIGURE 4. Horizontal Cross Section of a Hull at the Waterline

In Eq. (8), the minus sign is used if G is above B_o ; the plus sign is used if G is below B_o .

Requirements for stability indicate the desirability of having a large value of metacentric height. On the other hand, results of model tests show that a large metacentric height promotes rapid rolling of short period, and this imposes relatively heavy stresses on a ship's structure, particularly those parts of the structure far removed from the axis of roll. The rapid rolling also causes discomfort to both crew and passengers. In addition, a high metacentric height tends to produce greater amplitudes of roll from synchronous waves. Therefore, ocean-going vessels tend to have a lower metacentric height than vessels designed for use on inland waters.

In the interests of safety, the United States Coast Guard[9] regulates the minimum value for passenger vessels. The criteria for regulation are based on such factors as wind load, passengers, and damaged-ship operation. Military vehicles should be designed with these regulations in mind. Under Coast Guard regulations, the required minimum metacentric height (in feet) for consideration of weather (wind load) is

$$\overline{GM} = \frac{PAh}{\Delta \tan \theta} \tag{9}$$

where $P = 0.005 + \left(\frac{L}{14,200}\right)^2$ for ocean and coastwise service (L is the waterline length at maximum loading, in feet)

$P = 0.0033 + \left(\frac{L}{14,200}\right)^2$ for partially protected waters such as lakes, bays and sounds, and Great Lakes (summer service)

$P = 0.0025 + \left(\frac{L}{14,200}\right)^2$ for protected waters such as rivers, harbors, etc.

A = projected lateral area of that portion above the waterline, sq ft

h = vertical distance from the center of A to the center of the underwater lateral area (approximately the one-half draft point), ft

Δ = displacement, long tons

θ = angle of roll to one-half the freeboard or 14 degrees, whichever is less

The required minimum metacentric height for consideration of passengers is obtained from

$$\overline{GM} = \frac{Nb}{24\Delta \tan \theta} \tag{10}$$

where N = number of passengers

b = distance from the vessel's centerline to the geometrical center of the passenger-deck area on one side of the centerline, ft

Δ = displacement, long tons

θ = angle of heel to one-half the freeboard or 14 degrees, whichever is less

The regulations covering a damaged-ship operation define extent of damage, relating it to ship category. Ship damage results in the flooding of compartments, and the regulations specify that provisions be incorporated in the ship design to ensure a symmetrical flooding which will minimize roll. The maximum roll angle is specified as 7 degrees or, by special permission of the U. S. Coast Guard, up to 15 degrees. The regulations specify that, regardless of symmetrical or unsymmetrical flooding, a positive residual metacentric height of at least 2 inches shall remain (calculation is based on initial undamaged displacement).

Typical values of metacentric height for undamaged vessels are shown in the table on page 14. Included in the table is the calm-water roll period t , beam, and displacement. A general, recommended metacentric height for most military floaters under most conditions should be near 2 feet.

The actual metacentric height varies with loading, both static (cargo) and dynamic (passengers and crew). The lower the cargo is placed, the lower the center of gravity and the greater the metacentric height. To

REPRESENTATIVE CHARACTERISTICS OF VARIOUS VESSELS[3]

Name of Ship	Type	t (sec)	Beam (ft)	$\overline{GM}$ (ft)	Δ (tons)
AMERICA	Passenger	28.00	74.30	1.40	27,500
MT. VERNON	Passenger	20.40	72.20	2.50	27,250
COVINGTON	Passenger	18.30	65.40	2.50	22,000
CONTE DI SAVOIA	Passenger	26.00	96.00	3.14	41,200
SALT LAKE CITY	Cruiser	12.10	64.00	5.55	11,500
OMAHA	Cruiser	17.00	55.30	2.00	8,300
OREGON	Coast-Defense Battleship	15.66	69.25	3.00	9,800
REVENGE (with no bilge keels)	Battleship	15.20	75.00	3.78	14,300
REVENGE (with bilge keels)	Battleship	16.80	75.00	3.29	13,370
DUKW	$2\frac{1}{2}$-ton 6x6 Amphibian	2.5	8	2.08	10
MTV(XM759)	$1\frac{1}{2}$-ton Air-Bag Amphibian	8.6(?)	9.15	-	6.5
LVTP-5	Tracked Amphibian	2.6	11.66	1.47	41.25
LARC V	5-Ton 4x4 Amphibian	3.3	9.7	1.80	15.5
SUPERDUCK	5-Ton 6x6 Amphibian	2.8	8.22	2.35	13.2
DRAKE	8-Ton 8x8 Amphibian	2.7	10.0	2.45	25

improve stability, all heavy loads should, therefore, always be placed at the bottom. The wider the vehicle and the lower its center of gravity with respect to the center of buoyancy, the greater will be the metacentric height.

In practice, it should also be considered that the consumption of fuel and the loading or unloading of cargo alter the vehicle's displacement and cause a resulting shift in the center of buoyancy and the center of gravity. In consequence, the initial metacentric height is also changed. The crew must be trained to distribute the loads in such a manner as to keep the metacentric height in a specified range while afloat. An example of a loading chart, which was developed for the $2\frac{1}{2}$-ton DUKW, is shown in Fig. 5. Similar charts should be developed for all operational amphibian vehicles, prior to their introduction into service.

The metacentric height may be determined experimentally. To do this, the vehicle, manned to its operating weight, is inclined in calm water (Fig. 6), by the shifting of a known load W a distance x from one side of the vehicle to the other. This transfer of load is equivalent to the application of a listing moment to the floating vehicle:

$$T_{listing} = Wx \cos\theta \tag{11}$$

In a state of equilibrium, the listing moment equals the righting moment.

$$T_{listing} = \beta \tag{12}$$

Or, from Eq. (7),

$$Wx \cos\theta = \Delta\, \overline{GM} \sin\theta \tag{13}$$

and

$$\overline{GM} = \frac{Wx}{\Delta \tan\theta} \tag{14}$$

Knowing the values W, x, and Δ, and measuring the vehicle's angle of list, θ, it is possible to determine the metacentric height $\overline{GM}$.

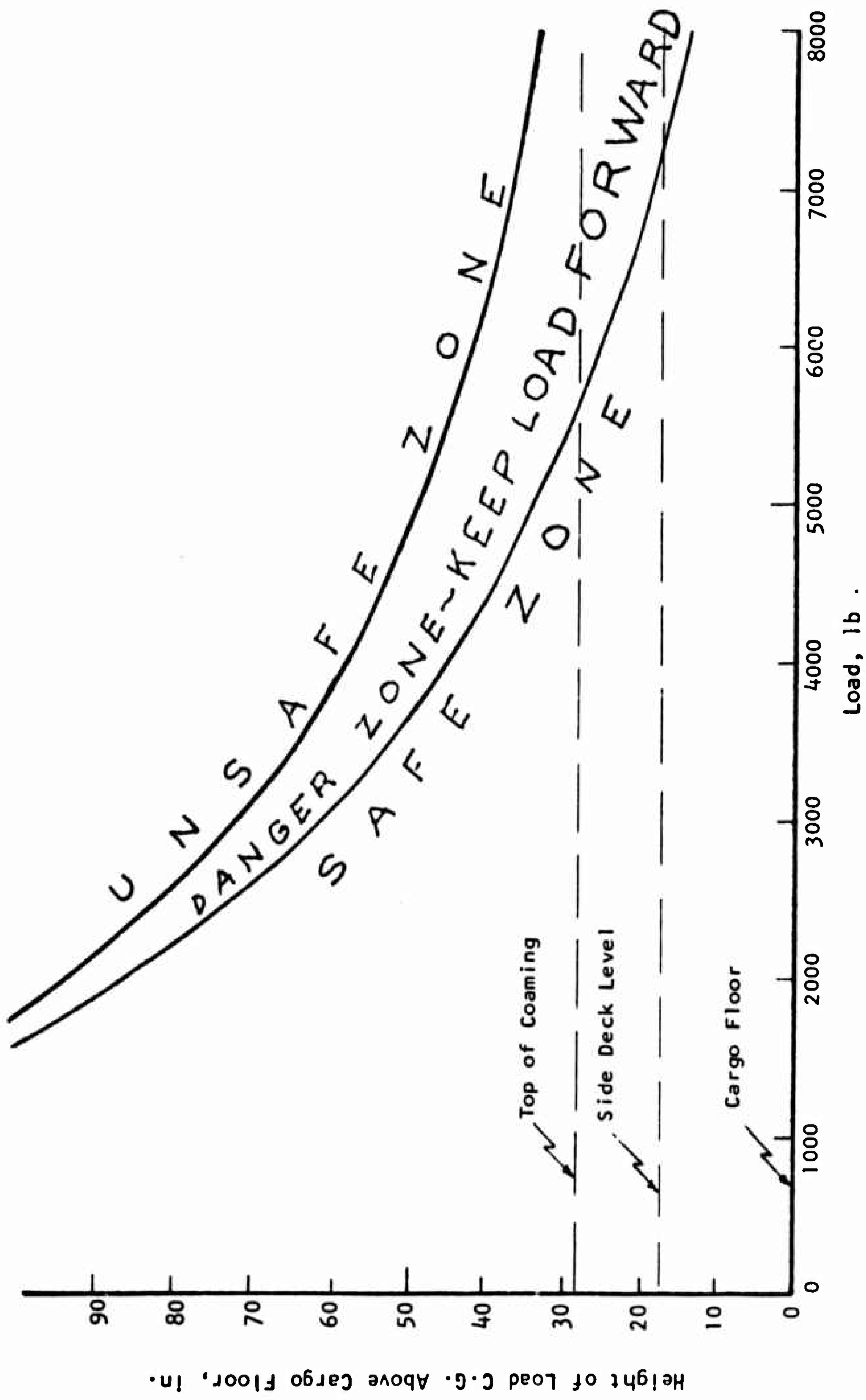


FIGURE 5. Loading Guide for DUKW Amphibian

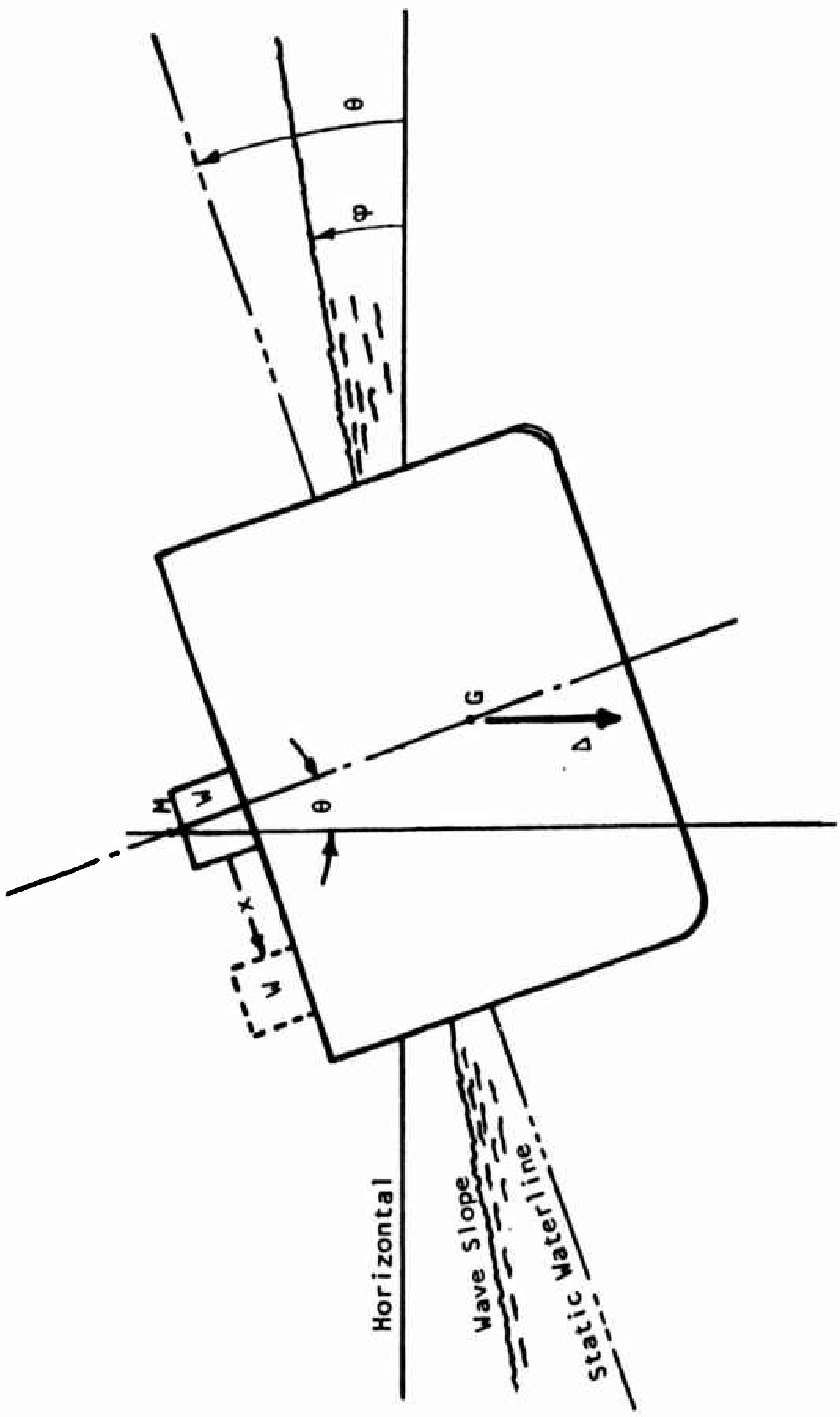


FIGURE 6. Cross Section of Ship Rolling in Waves

The distance between the metacenter and the center of buoyancy during listing is called the transverse metacentric radius; it also serves to determine the magnitude of vehicle stability. The transverse metacentric radius corresponding to a given waterline and a given angle of inclination is calculated by

$$r = \frac{I}{V} \tag{15}$$

where I = the moment of inertia of the water-plane area

V = the volume of displaced water which, of course, can be computed from the vehicle weight

An increase in r depends on an increase of I or a decrease of V. Decrease in the value V is strictly limited by the load requirement for the vehicle, so that the ratio I/V can be higher only with an increase in the moment of inertia of the water-plane area which depends on the hull shape. For a rectangular shape with a length ℓ and width b, for example, the moment of inertia is given by the formula $I = \ell b^3/12$, the width at the waterline being raised to the third power, so that for a fixed length the width to a large extent determines the vehicle's stability.

A liquid (fuel) cargo which completely fills the tanks does not affect the vehicle's stability. But if the liquid only partly fills the tanks and can flow freely during listing (that is, the shape of the liquid volume changes), then the stability of the vehicle always decreases. It should be noted that during listing the free surface of liquid cargo tends to assume a position parallel to the actual waterline. As a result, the center of gravity of the cargo moves in the direction of listing, shifting the center of gravity of the entire vehicle that way. The stability lever, thus decreased, reduces the vehicle's stability.

While keeping in mind the negative effect which liquid cargo has on stability, we must recognize the special importance of taking into account the presence of any sizable quantities of water in the bilge of the hull, resulting from seepage, rainfall, or other cause. Water in the bilge will slosh and can produce a decrease in stability. To minimize the effects of liquid cargos on stability, two fuel tanks should be installed in amphibious

vehicles, or a special longitudinal bulkhead should be installed when a single tank is used. If a tank is divided in half by such a single partition, the moment of inertia of the free surface for both of the compartments so formed is one-fourth of the moment of inertia of the free surface for the undivided tank. To reduce the effect of liquid cargos on stability, fuel tanks and compartments containing other liquids should usually be divided into sections with the least possible width.

In both the design and operation of amphibious vehicles, care must be taken to avoid the loss of stability which would result from swamping or capsizing. In the smaller vehicles, particular care must be taken to avoid "rocking the boat." The movement of crew on board a vehicle shifts the center of gravity, and hence promotes rolling. Standing up in a boat raises the center of gravity, and hence reduces the metacentric height.

Longitudinal stability, which minimizes the heaving of the vehicle, is usually considerably greater than transverse stability, and does not require special attention.

ROLLING AND PITCHING MOTIONS

In calm water the period of roll (or trim) may be obtained from[3]

$$t = \frac{1.108\ k}{\sqrt{\overline{GM}}} \tag{16}$$

where t = period of roll, expressed in seconds

$\overline{GM}$ = metacentric height, expressed in feet

k = radius of gyration about the roll axis, approximately equal to 0.4 times the beam; expressed in feet

The value of this period is significant for the following important reasons:

(1) As mentioned earlier, a period which is too short will cause excessive strain to be placed on the vehicle's structure, and the crew will get a "jerky" ride.

(2) A period which is too long will cause excessive angular roll to take place, with a corresponding inducement of motion sickness among crew and passengers.

(3) A period which is synchronous to the waves will cause sympathetic impacts to take place; these will induce extremely high roll angles and lead to the possibility of capsizing. The table on page 14 contains representative roll periods which are acceptable for various vehicles.

The oscillating motion of a vehicle in waves is caused by the action of external forces while the vehicle is moving on the surface of either calm or rough water. Oscillations arise from the action of waves as their crests produce additional buoyant forces first on one side and then on the other side of the hull, developing moments which cause heel or trim. The way a vehicle behaves on rough water depends both on the waves striking the vehicle and on the shape of the vehicle's hull. The oscillations may be transverse (rolling), longitudinal (pitching), or vertical (heaving). During rolling the vehicle executes a rocking motion around a longitudinal axis, but during pitching it rotates around a transverse axis.

Mixed oscillations also occur when the vehicle is subjected to several types of motion simultaneously. The combined pitch and heave motions usually cause a vehicle to pitch about an axis located approximately two-thirds aft of the bow and near the static water plane, and to roll approximately midway between the waterline and the height of the center of gravity. Any weapon mountings should therefore be placed as close to this location as possible, if to be employed during swimming.

Rolling and pitching impairs the navigational characteristics of the vehicle and increases the loads on the hull from added forces, water resistance, and wave impacts. It also lowers the vehicle speed, increases fuel consumption, and causes water to break over the bow and stern. Rolling may also lead to loss of stability, capsizing, and impairment of the crew's efficiency.

Although the water in a wave appears to move forward, the water particles actually move in a near circular path (elliptical in shallow water), with but a slight forward motion (drift) in the direction of the

wave. Waves are characterized by three basic elements: height, length, and period. The height of a wave is the elevation from the crest to the trough (along the vertical); the length of a wave is the distance between two adjacent crests or troughs; and the period of a wave is the time needed by a particle of liquid to perform one complete oscillation. The relationship of wavelength (λ) and period (t_w) for water waves is defined by

$$\lambda = \frac{g}{2\pi} t_w^2 = 5.12\, t_w^2 \qquad (17)$$

The height of wind-generated waves depends on the force of the wind as well as on the depth and length of the stretch of water. On rivers, wave heights up to 1 to 1.5 feet are the maximum to be expected. On lakes, heights of 3 feet and higher have often been encountered. In open seas, waves of 60 feet and higher are encountered in the most violent storms; but amphibious operations are rarely conducted in waters rougher than Sea State 3 (roughly equivalent to the condition existing when the 1/3 highest wave height is 4.6 feet; two-thirds of the waves exhibit a value less than this).

There is no definitive relationship between wavelength and wave height. Open-sea waves usually break when they are steeper than 10:1 (slope about 12 deg). In surf, however, much greater ratios are encountered and 7:1 (16 deg) waves have been observed. No relationship has yet been established between the heights of waves in an open sea and the corresponding heights as waves break on a beach. Waves near shore have, on the whole, received little attention in study programs, and not too much is known of them at this time.

The rolling of a vehicle in a given regular-wave pattern can be readily calculated. Actual waves, however, are random in both length and height. Yet because the major energy of waves is concentrated within a rather narrow bank of wavelengths, the oscillations for a wavelength centered in that band may be employed to obtain a rough characterization. The equations presented here are for long waves whose length (λ) is large

compared to the beam of the vehicle. The equations were developed to determine the angle of vehicle roll, as a function of the vehicle and wave properties. The phenomenon of synchronism between wave period t_w and vehicle period t is presented, and the effect of synchronism on maximum roll angle is shown.

Figure 6 (page 17) is a cross section of a floating body rolled to an angle θ from the horizontal, on the slope φ of a long wave. For this body, the righting moment, from Eq. (7), is

$$\beta = \Delta \overline{GM} \sin \theta$$

The wave surface is assumed to be near a sine wave in shape; hence,

$$\varphi_i = \varphi \sin \frac{2\pi\tau}{t_w} \tag{18}$$

where φ = maximum wave slope

τ = time

t_w = period of the wave

φ_i = instantaneous wave slope

The equation of motion for rolling may be derived as

$$\frac{4\pi^2}{t^2} \left(\theta - \varphi \sin \frac{2\pi\tau}{t_w}\right) + \frac{4k_d}{t} \left(\frac{d\theta}{d\tau}\right) + \frac{d^2\theta}{d\tau^2} = 0 \tag{19}$$

where

$\frac{4k_d}{t} \left(\frac{d\theta}{d\tau}\right)$ = resistance to rolling

k_d = damping coefficient

t = the natural period of vehicle roll

The solution of Eq. (19) is

$$\theta = \frac{\varphi \sin\left[(2\pi\tau/t_w) - \alpha\right]}{\left\{\left[1 - (t^2/t_w^{\,2})\right]^2 + (4k_d^{\,2}/\pi^2)(t^2/t_w^{\,2})\right\}^{0.5}}$$

$$+ \theta_{max} \sin\left(\frac{2\pi\tau}{t} \sqrt{1 - (k_d^{\,2}/\pi^2)} + \theta_o\right) \exp\left[- \frac{2k_d\tau}{t}\right] \qquad (20)$$

where

$$\alpha = \tan^{-1} \frac{2k_d}{\pi} \left[\frac{t/t_w}{(1 - t^2/t_w^{\,2})^2}\right]$$

θ_{max} = maximum roll angle

θ_o = roll angle at time equals zero

The second term in Eq. (20) diminishes as time increases and is usually neglected, except in cases of unusual initial conditions such as large roll angle.

The first term of Eq. (19) is the forced oscillation conforming to the period of the waves. Therefore, after a short time, the roll (θ) will be a maximum when the forced oscillation is a maximum, or when

$$\sin\left(\frac{2\pi\tau}{t_w} - \alpha\right) = \pm 1 \qquad (21)$$

Therefore the maximum roll angle (θ_{max}) in waves is

$$\theta_{max} = \frac{\varphi}{\left\{\left[1 - (t^2/t_w^{\,2})\right]^2 + (4k_d^{\,2}/\pi^2)(t^2/t_w^{\,2})\right\}^{0.5}} \qquad (22)$$

Synchronism is defined as the condition existing when the wave period (t_w) equals the floating-body period (t). At synchronism, the maximum roll (θ_{max}) is limited only by the damping coefficient (k_d). In the case of ships, the damping coefficients vary from values of 0.01 up to 0.09, depending on roundness of bottom, presence of bilge keels, hull roughness, width of beam, etc. In the case of amphibious vehicles, the damping coefficient (k_d) will be large, because of the wheels or tracks, differentials, housings, suspension systems, rub rails, etc., protruding from the hulls.

The damping coefficient k_d can readily be determined for an existing amphibian by application of the second term in Eq. (20) to a roll test in calm water. The maximum roll angle will decay with time, and the decay is attributable to the damping coefficient. If the damping coefficient of an amphibian is not known, a value of 0.10 to 0.15 is reasonable for a first-cut estimation.

The effect of synchronism on the amplitude of forced rolling is illustrated in Fig. 7, where the ratio of maximum roll angle to maximum wave slope is plotted against the ratio of ship and wave periods. It can be seen that, for values of t/t_w between 0.8 and 1.4, the forced inclination exceeds twice the wave slope. The wave slope seldom exceeds 9 degrees. Hence, if the ratio t/t_w is kept outside this range, dangerous rolling is not to be anticipated. We may be guided by the fact that the wave periods with greatest energy in open seas are between 5 and 30 seconds; in inland waterways, they are between 2 and 6 seconds. Since the free period of rolling of most amphibious vehicles is usually from 2 to 4 seconds, these vehicles may frequently be found in synchronism with the waves, in which case roll can become very dangerous.

If we consider two vehicles, one with a large and the other with a small righting moment, the latter vehicle will, after listing, return to its upright position more slowly than the first. The period of free rolling for the second vehicle is therefore greater than for the first. When both vehicles are tilting to the same angle, the oscillations for the second vehicle (with small righting moment) are smooth, but those for the first vehicle (with large righting moment) are precipitous.

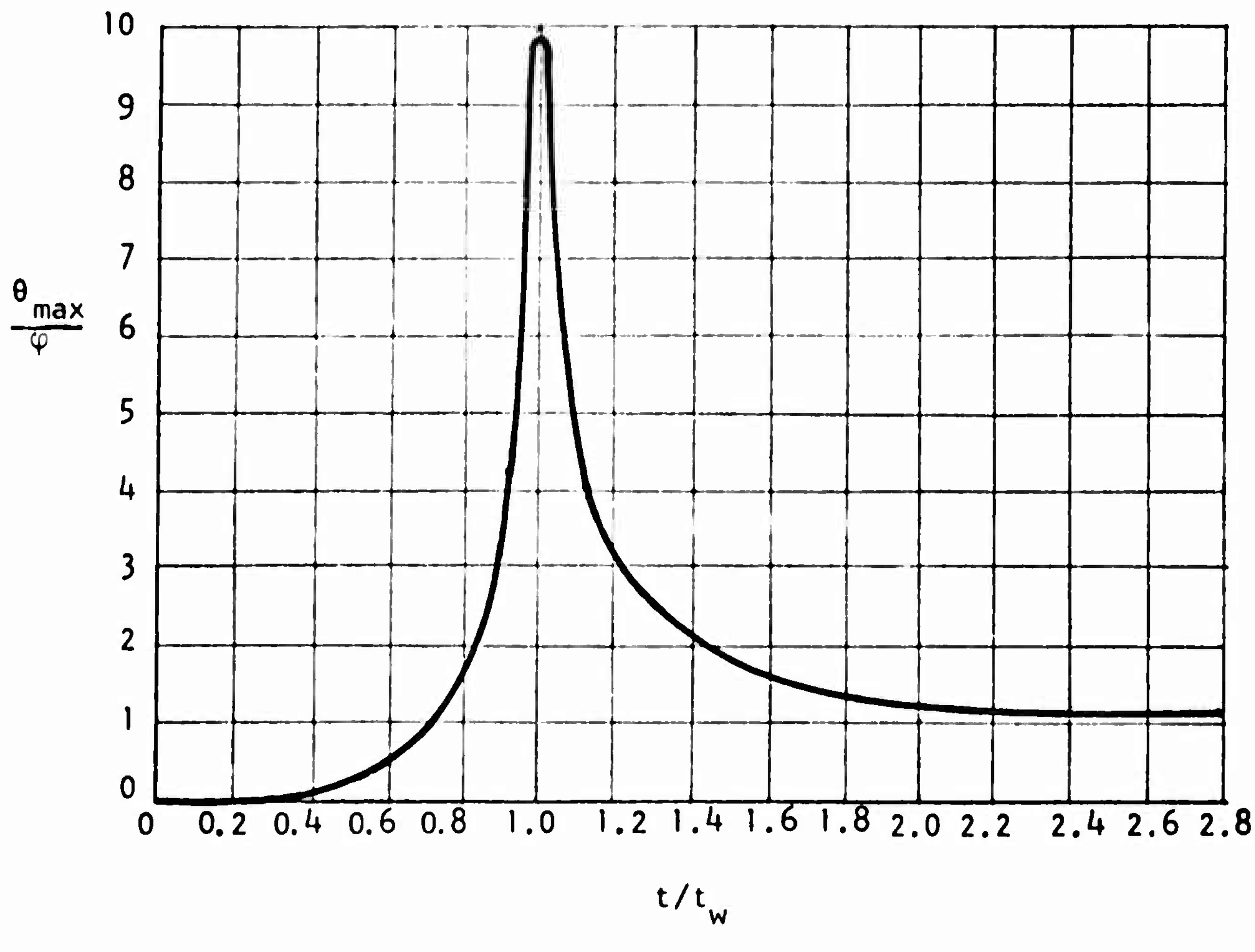


FIGURE 7. Amplitude of Forced Rolling for Various Values of Roll-Wave Period Ratio

To decrease the rolling of ships, special devices such as gyroscopic stabilizers and bilge keels are used to restrict the amplitude of rolling. Tanks located along the sides of the ship and transversely connected with pipes are also used. With these tanks, the effect of waves can be diminished by pumping water or any other liquid from the tank on one side to a tank on the other, in a direction opposite to the action of the waves. Specially designed tanks can achieve out-of-phase motion without pumps, and should be considered in any design if practical.[3]

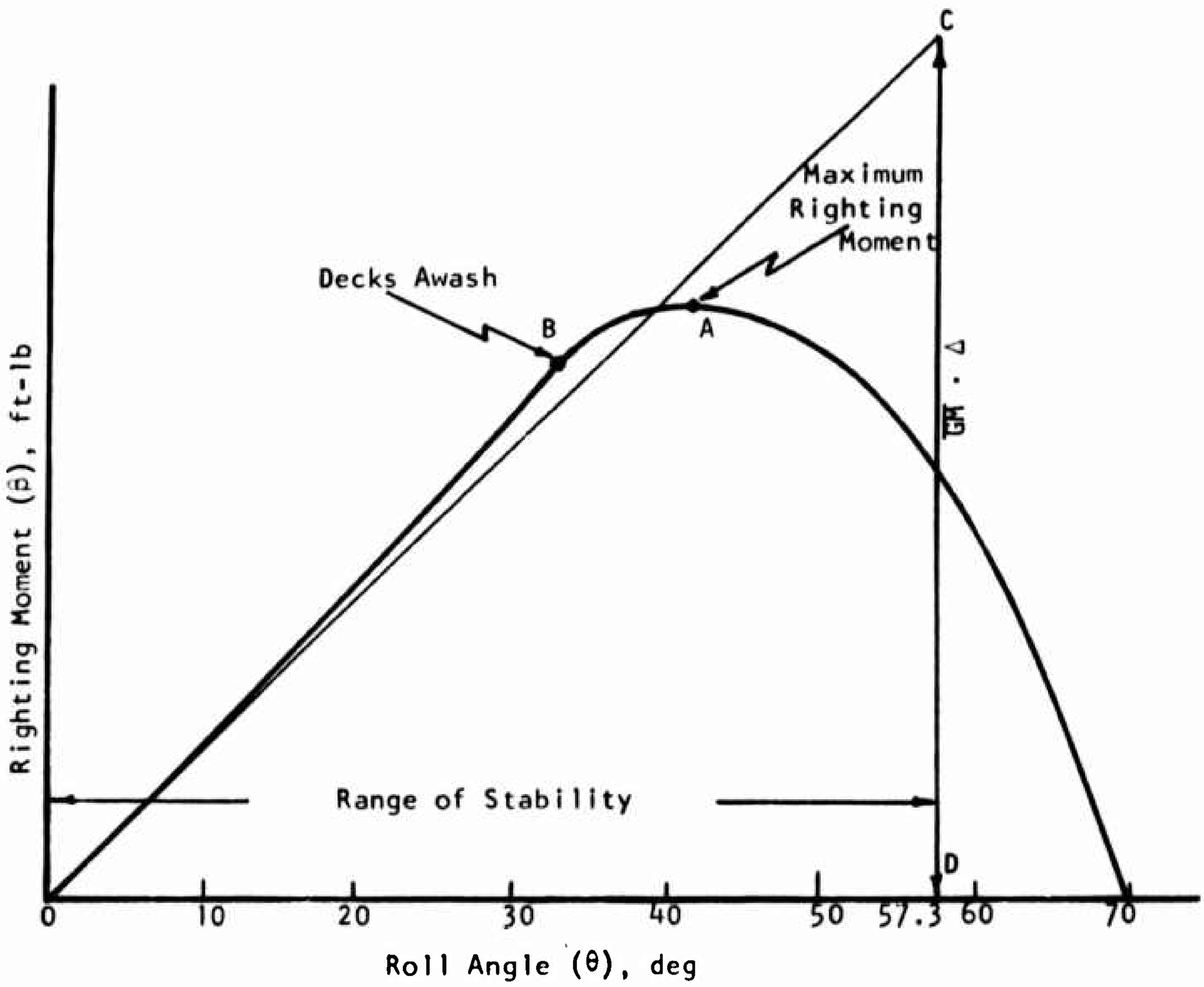
Rolling can also be reduced merely by changing the direction or speed of the vehicle, with respect to the oncoming waves. Basically, this affects the period of the disturbing force generated by the wave action and the magnitude of rolling.

RESISTANCE TO CAPSIZING

A plot of the righting moment (β) versus roll angle (θ) will yield the curve of static stability (see Fig. 8). This curve has a number of features which are significant in analysis of the vehicle's stability. The apex of the curve (A) gives the value of the maximum righting moment, and the angle at which it occurs.

The point of deck-edge immersion occurs at point B, where the curve changes direction. The curve presented in Fig. 8 shows the range of stability from zero to 70 degrees (which is the angle of vanishing stability at which the vehicle will capsize). Note the shape of the curve at the origin. It will always be tangential to the line OC, where CD, erected at 57.3 degrees (one radian), has the value $\overline{GM} \cdot \Delta$.

The angle for entry of water, and consequently the maximum stable angle of list, depends on the height of the freeboard of the vehicle. The shape of the static stability curve is different for every vehicle. The steeper the slope of the curve, the greater the initial metacentric height and the faster the righting moment rises with increasing vehicle roll. But a large initial stability, characterizing a large initial transverse metacentric height, does not always mean that the vehicle will be very stable at large angles of list. Curve I of Fig. 9, for example, has a steep



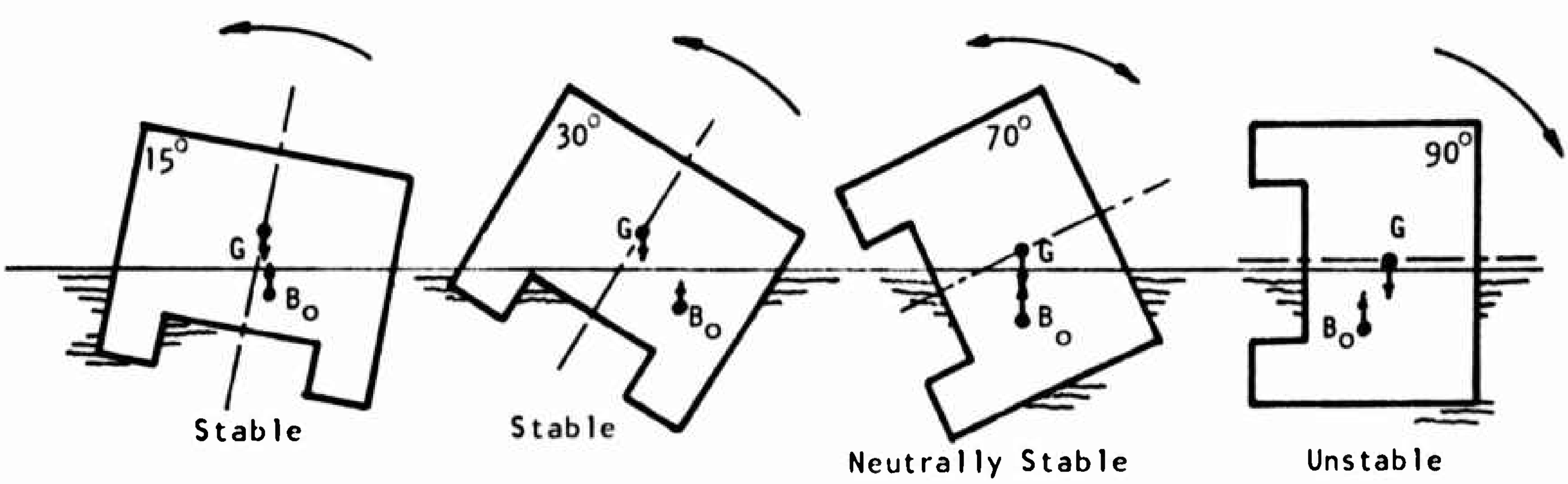


FIGURE 8. Diagram of Static Stability

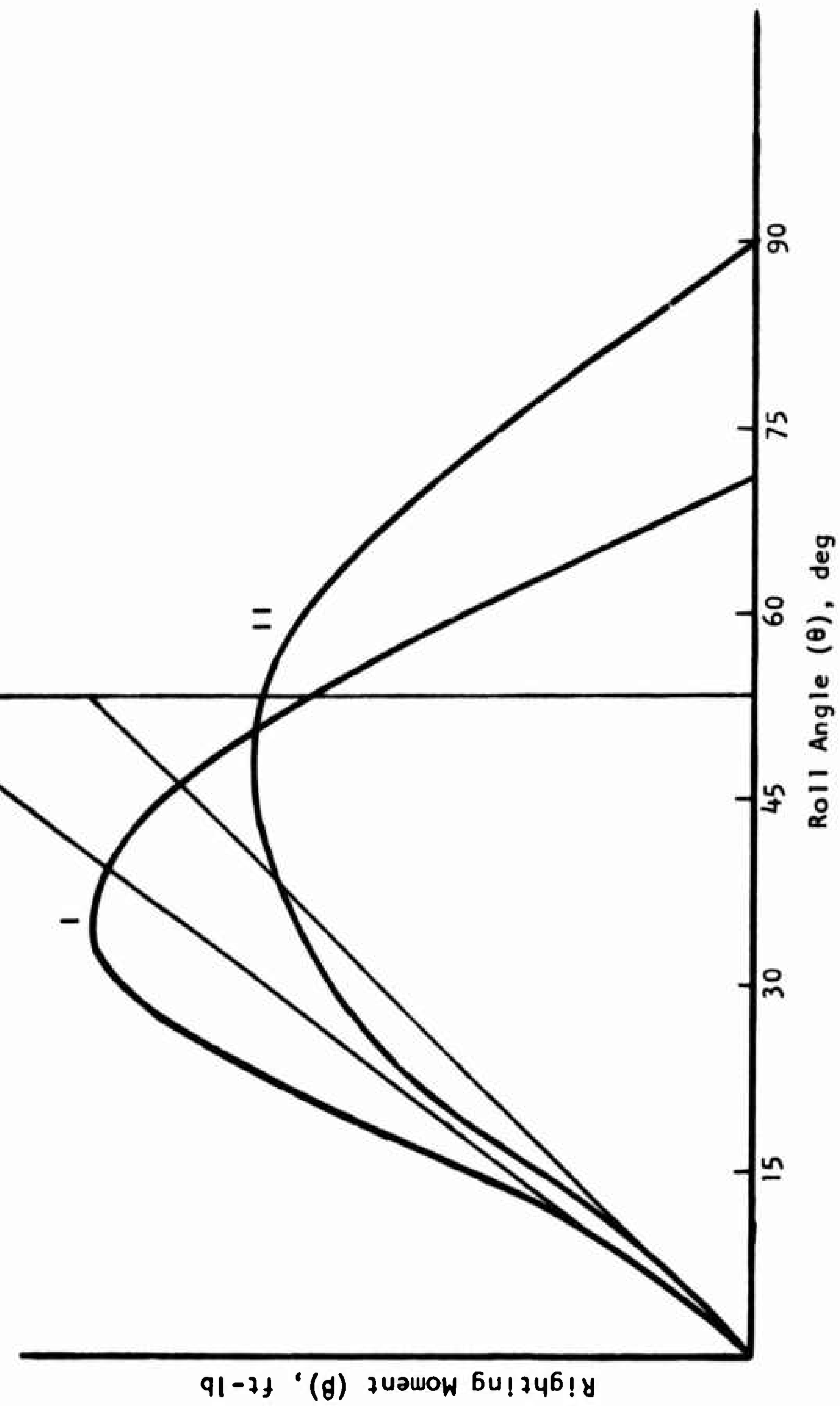


FIGURE 9. Static Stability for Vehicles of Different Type

initial slope; it quickly reaches a maximum, and then drops off quickly; that is, the vehicle loses stability. Curve I pertains to a vehicle with a rather wide hull but a low freeboard. Curve II has a lower initial metacentric height, but its peak is reached at a larger angle of list. This denotes good stability at large angles of inclination. Curve II characterizes a vehicle with deep draft and high freeboard.

During use, a vehicle is acted on by a variety of dynamic forces, that is, the action of waves, the forces generated during turns, the changes in loading, the tensile forces in tow lines, and the impacts of underwater obstacles or floating objects. In these cases, the tipping and righting moments are initially unequal; the amphibious vehicle starts to list and exhibits an angular acceleration (see Fig. 10). On reaching the angle at which the righting moment equals the upsetting moment, the vehicle achieves its maximum angular velocity and will continue to incline further, but at a decreasing rate. The vehicle will stop tilting when the work done by the upsetting moment becomes equal to the work of the righting moment (minus dynamic damping). If the upsetting moment is transitory, the vehicle will stop for an instant, then return to its normal position. If the upsetting moment remains, the vehicle will arrive at a state of rest which will be determined by the angle of list whose righting moment equals the applied upsetting moment (point B in Fig. 10). Therefore, to determine the maximum angle of list for a dynamically applied upsetting moment, it is necessary to locate, on the static-stability curve, a point for which the work of the upsetting moment (area ABO) equals that of the righting moment (area BDC).

Examination of the earlier diagrams of static stability for different vehicles warrants the conclusion that the greater the ordinate of the static-stability diagram and the greater the area under the curve (i.e., the farther the distance from the origin to the point of intersection of the curve with the abscissa) then the greater the vehicle stability. The area of the diagram depends, basically, on the vehicle's freeboard. The higher the freeboard, the greater the stability at large angles of list, and the greater the protection against overturning.

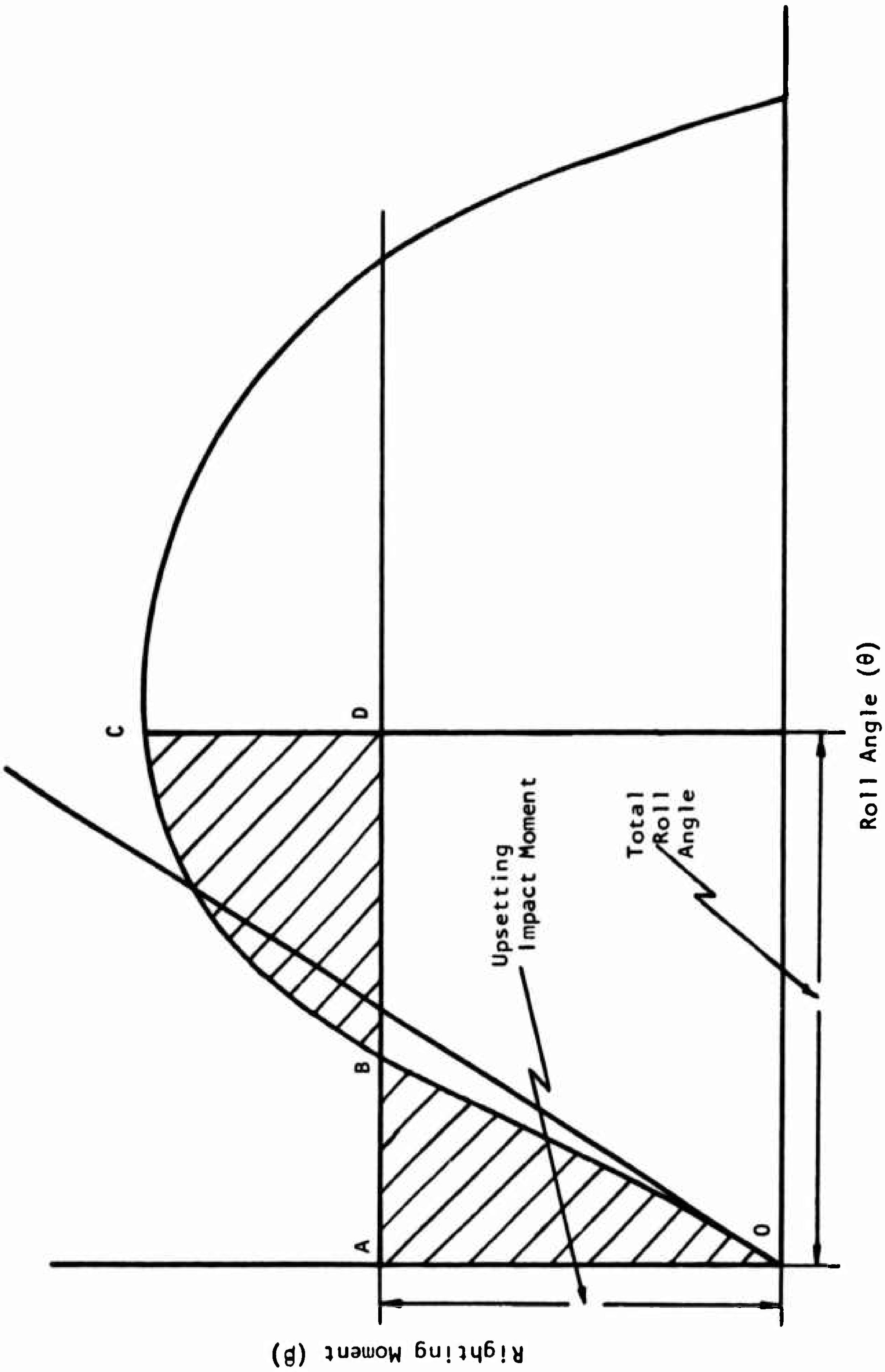


FIGURE 10. Stability Determination for Dynamic Loads

FREEBOARD CONSIDERATIONS

Reserve Buoyancy

If the vehicle carries more cargo than anticipated it will sink deeper into the water but it will not lose buoyancy until water runs over the coamings into the hull. The watertight volume of the hull, located above the load waterline, is called the reserve buoyancy. Reserve buoyancy enables a vehicle to accept more than the assigned load and still remain afloat. It includes, in addition to the main hull, any additional watertight superstructure or volume-displacing equipment that is external to the hull and becomes submerged before flooding occurs. A vehicle's auxiliary watertight above-water volumes can compensate for lost supporting force during an unexpected increase of immersion resulting either from the taking on of cargo or from damage to below-water portions of the hull.

Vehicle freeboard is the most common form of reserve buoyancy. But the freeboard at the same time represents a negative factor, since increase in freeboard raises the vehicle's center of gravity and increases its silhouette, weight,and exposed sail area.

There is a relative scarcity of existing data on which to base water freeboard requirements for military vehicles. The requirement for freeboard is actually an effort to define the roughness of the water in which the vehicle can operate. Current requirements were apparently established on the basis of somewhat limited experience with earlier vehicles of similar type.[10] The required reserve buoyancy, and consequently the needed freeboard, depends on the dimensions of the vehicle, its mission, the strength of its hull, its self-bailing capacity, and many other factors. It is expressed as a percentage of the normal displacement and should be in the neighborhood of 20 percent or greater for amphibious vehicles. Some amphibians, notably the LARC V, the M113,and the LVTP-5, have very low freeboard, but have instead a watertight deck (wet deck) which seals the interior against water entry. Such vehicles may be totally submerged, temporarily (as in a heavy surf), without swamping.

Swamping

In design studies of small boats and amphibians, it is important to consider susceptibility to swamping. Generally speaking, swamping conditions can arise in any one of certain circumstances:

(1) During ingress at a steep angle, parts of the bow may submerge before sufficient buoyancy is developed to maintain freeboard.

(2) During egress, parts of the stern may be forced under while the forward parts of the vehicle are negotiating the bank.

(3) Under rough-water conditions, waves may enter the vehicle over the gunwales.

(4) The bow wave (static water rise) produced by a blunt-shaped bow may be high enough to enter the vehicle.

(5) If the vehicle is restrained in a strong current (perhaps when hung up on a submerged obstacle or when attached to a line used as an aid in crossing), current impinging on the upstream side may generate a static water rise capable of swamping the vehicle.

In addition to the water flow to either side of the vehicle, which can easily be seen by an observer, there is also flow under the bottom of the vehicle. This flow produces an increase in the speed of the water, relative to the vehicle, and a corresponding decrease in hydrodynamic pressure under the vehicle. Depending on freeboard and bow shape, a vehicle may pitch downward and become vulnerable to the danger of being swamped by its own bow wave. Vehicles which have a fair bow treatment, such as the DUKW with its barge-like hull or the LARC with its addition of a prow, can generate a lift force with the inclined surface such a bow presents to the flow. Such bow treatment generally reduces the hydrodynamic drag, but it helps most by reducing the swamping tendency. Placing the propulsion thrust line below the center of gravity will also produce a favorable bow-up moment.

To protect the vehicle, the designer has three basic alternatives. He may choose one, two, or all three:

(1) An increased freeboard which will lessen the chances of swamping and reduce the quantity of water entering the vehicle when shipping occurs.

(2) Enough watertight compartments to maintain buoyancy regardless of the amount of water entering ("self-bailing" and "wet deck" designs).

(3) Adequate pumping capacity, so that water can be removed from the vehicle at a rate exceeding the rate of inflow.

Shipping Rates

To establish freeboard criteria, to determine bilge-pump size, and to specify vehicle operational limitations, methods must first be developed for predicting water-shipping rates for the aspects of amphibious operations to which each problem is related. The established literature seems to be totally lacking in information on the prediction of the shipping rates of vessels.

Numerous model tests have been conducted to establish the performance of various designs in calm and rough water. The results, however, have been rather qualitative in nature, establishing forward speeds at which shipping occurs or at which the bow wave exceeds desirable heights.

Model tests have also been conducted[11] to establish the maximum bank angle that a given vehicle can negotiate without either bow or stern submerging. To determine the extent to which approximate calculations might be developed for the prediction of the rate of water shippage in a dynamic situation, a theoretical approach was postulated and a limited number of model tests were conducted to validate the equations (which are presented here in but a general form). The results of this effort appear in the following section.

Predictions of Shipping Flow

A. Ingress and Egress

During ingress and egress, the possibility exists that the coaming line will dip below the free water surface, permitting water to flow into the vehicle temporarily. In this case, the flow rate may be estimated on the basis of equations which predict the flow over a horizontal broad-crested weir. The following equation is well documented in the literature of fluid mechanics:

$$Q = 3.33 \; Lf^{3/2} \tag{23}$$

where Q = water flow rate, cfs

f = freeboard (negative), ft

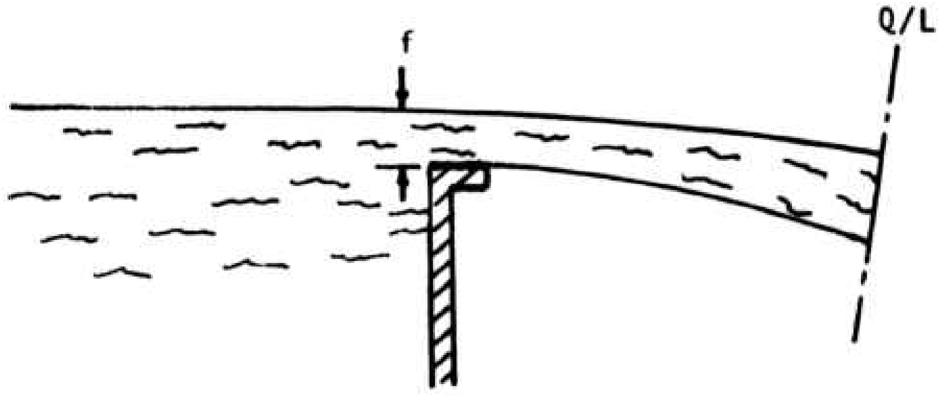
L = length of submerged coaming, ft

The coefficient 3.33 includes the empirical discharge coefficient and the necessary dimensional conversion factors. For ease of use, Eq. (23) has been plotted in Fig. 11 as negative freeboard in inches versus shipping rate in gallons per minute per foot of submerged freeboard.

In most real situations, the egress or ingress operation is transitory, hence the coaming is submerged only a discrete period of time. The total amount of water shipped, therefore, is a function of both the time required to conduct the maneuver and the depth of submergence.

B. Static Water Rise

Equation (23) may also be applied to calculation of the rate of water shippage when, in the course of a normal operation, a bow wave rises above the coaming line; and to the critical case of hang-up in a stream current. In these cases the negative freeboard (f) in Eq. (23) must be replaced by the water rise (z) due to the impinging water, minus the available freeboard. Obviously the equation is applicable only when the water rise exceeds the freeboard.



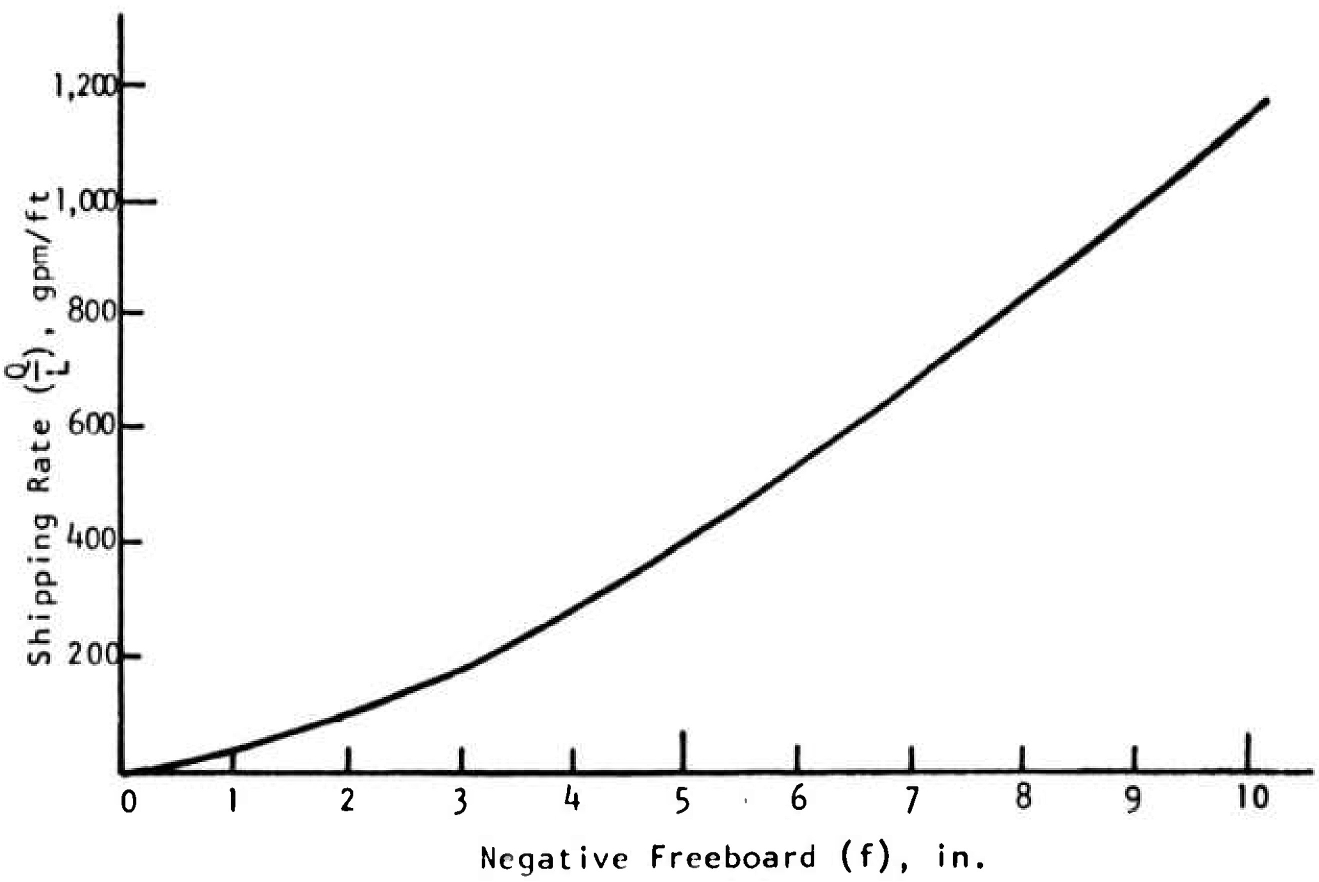


FIGURE 11. Rate of Water Shippage Over Submerged Coaming

Theoretically, the water rise (z) may be estimated by using the famous Bernoulli equation:

$$z = \frac{v^2}{2g} \qquad (24)$$

where v = velocity of the water with respect to the craft

g = gravitational constant

However, tests[12] indicate that the water flow below and around the vehicle tends to reduce this rise; the observed values are somewhat less than predicted by Eq. (24) and are also a function of vehicle draft. Several unsuccessful attempts have been made to determine the functional relation of this dependence. The general trend of the dependence, however, indicates that, as draft increases, actual water-rise approaches that predicted by Eq. (24). Figure 12 shows the results of some of the tests and may be used as a guide.

C. Waves

The dynamics associated with vehicle motion in waves are quite complex, and the prediction of general shipping rates in waves is clearly beyond the scope of this work. Within the present state of the art, such vehicle performance can be treated only with scale-model tests. It may be possible, however, to examine the effects of waves which can be high enough to reach beyond the coaming at certain points of a minimal-freeboard vehicle, but which are not great enough to cause any material pitching, heaving, or rolling of the body. Such analysis may also be useful in considering the effect of a few relatively large waves (such as can be generated by the passage of another vessel or by an explosion), which pass by the vehicle before the vehicle has time for measurable dynamic response.

The deep sea wave is usually depicted as a trochoidal curve -- a rather complex function. The wavelets considered here, are, for ease in computation, assumed to be well approximated by a simple sine wave such as

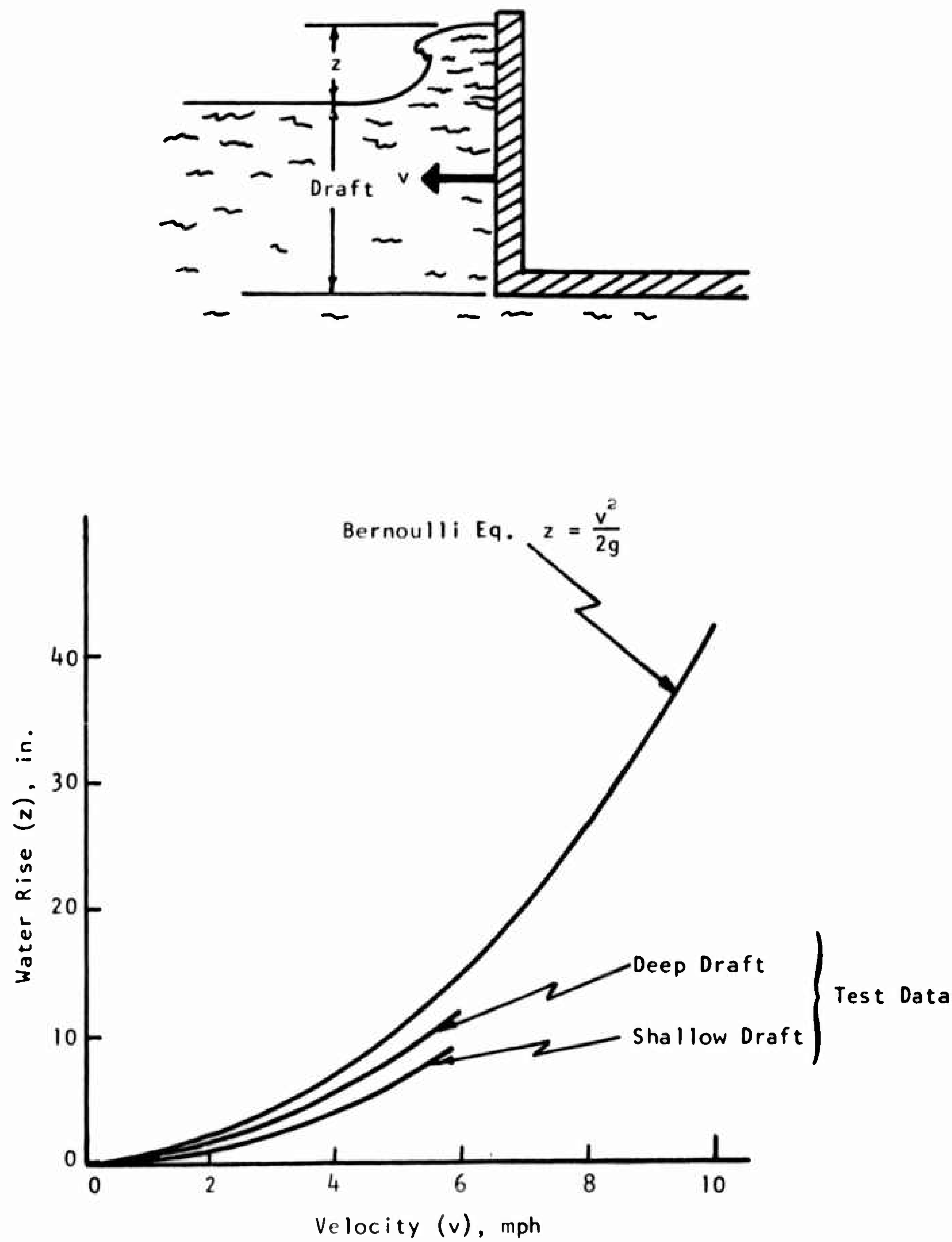


FIGURE 12. Bow Wave Height vs. Forward Velocity

that shown in Fig. 13. Further, the average flow rate over the coaming is assumed to be the instantaneous flow rate specified for a broad-crested weir by Eq. (23). For a given freeboard (f) from the still waterline, and an instantaneous wave height (y), the integrated flow rate over the length of the wave may be presented as

$$Q = 3.33L\,(y-f)^{3/2} \quad \text{for} \quad y - f > 0$$
$$Q = 0 \quad \text{for} \quad y - f \le 0 \tag{25}$$

Let $y = \frac{h}{2} \sin \omega$, where h is the crest-to-trough wavelet height (see Fig. 12); then

$$Q_{ave} = \frac{1}{2\pi} \int_0^{2\pi} Q\, d\omega$$
$$= \frac{1}{2\pi} \int_0^{\omega_1} Q d\omega + \frac{1}{2\pi} \int_{\omega_1}^{\omega_2} Q d\omega + \int_{\theta_2}^{2\pi} Q d\omega$$
$$= \frac{1}{2\pi} \int_{\omega_1}^{\omega_2} Q d\omega \tag{26}$$

since $Q = 0$ in the intervals from 0 to ω_1 and from ω_2 to 2π .

Now at ω_1 and ω_2 , y is equal to f ; therefore

$$f = (\tfrac{h}{2}) \sin \omega_1 = (\tfrac{h}{2}) \sin \omega_2$$

Solving for ω_1 and ω_2 , ω_1 is the value of $\sin^{-1}(2/h)$ between 0 and $\pi/2$; ω_2 is its value between $\pi/2$ and π . Therefore,

$$Q_{ave} = \frac{1}{2\pi} \int_{\omega_1}^{\omega_2} (y - f)\, d\omega = \frac{1}{\pi} \int_{\omega_1}^{\pi/2} 3.33L\,(\tfrac{h}{2} \sin \omega - f)^{3/2}\, d\omega$$

$$0 < \omega_1 \le \frac{\pi}{2} \tag{27}$$

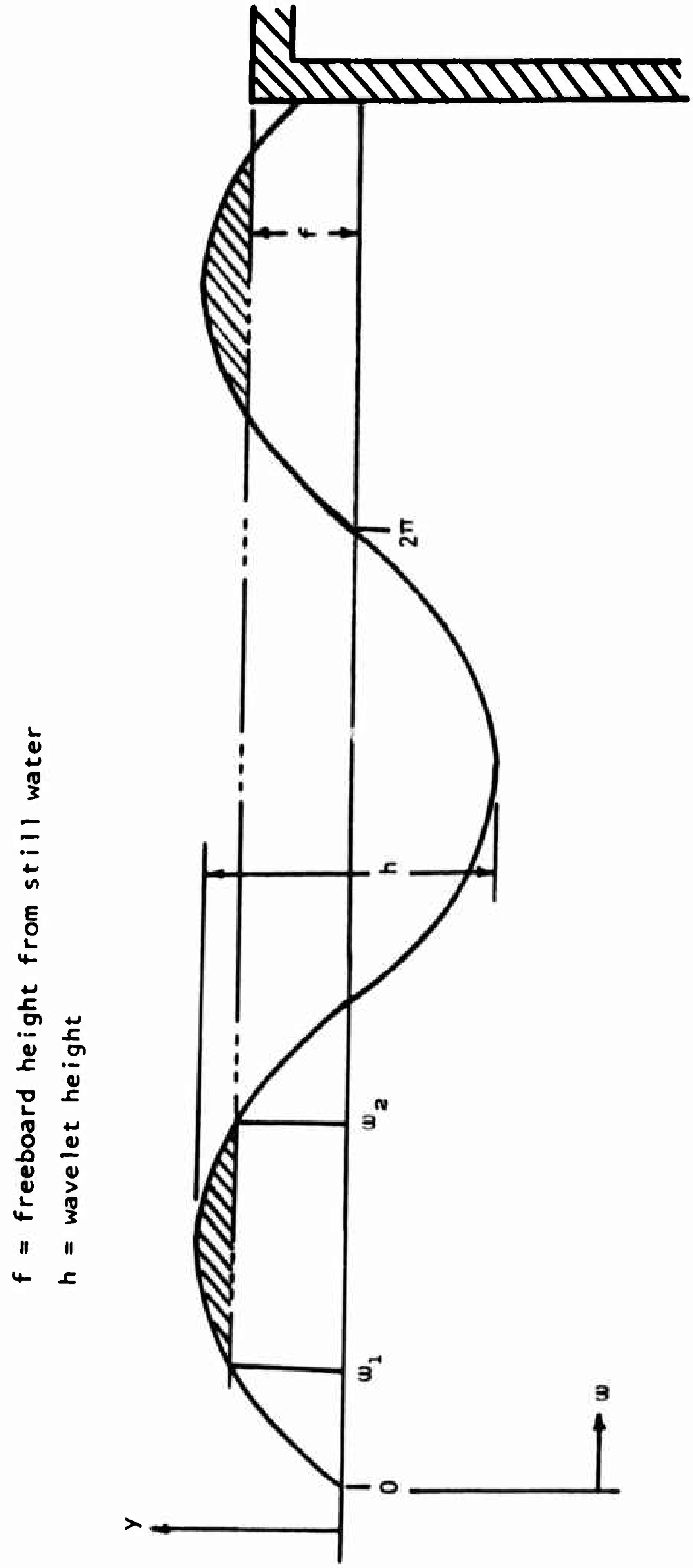


FIGURE 13. Sine-Wave Model of Wavelet

Equation (27) then becomes

$$Q_{ave} = \frac{3.33L}{\pi} \left(\frac{h}{2}\right)^{3/2} \int_{\omega_1}^{\pi/2} \left(\sin\omega - \frac{2f}{h}\right)^{3/2} d\omega \tag{28}$$

Consider the integral in Eq. (28). Let

$$\Phi = \int_{\omega_1}^{\pi/2} \left(\sin\omega - \frac{2f}{h}\right)^{3/2} d\omega$$

This integral has no closed-form solution. Expanding by a Taylor series and integrating the first six terms yields values of Φ which are plotted versus $\frac{2f}{h}$ as the solid line on Fig. 14.

Thus, for any known freeboard (f), wave height (h), and length of coaming (L), the rate of water-shipping should be approximated by the formula

$$Q_{ave} = \frac{3.33L}{\pi} \left(\frac{h}{2}\right)^{3/2} \Phi \tag{29}$$

where Φ is obtained from Fig. 14.

Tests were conducted at the Davidson Laboratory's Tank 3, to check Eq. (26). Worden[13] has given a detailed description of these tests and their results. It was found that considerably more water entered the vehicle than would be predicted by Eq. (29). (See plotted points on Fig. 14.) In fact, water entered the model when h was less than 2f.

Water particles in a wave move in a somewhat circular fashion, not in the idealized upward-downward motion used to generate Eq. (29). There exists, therefore, a component of the velocity (of those water particles) which is normal to the wave front and which at the top of the wave almost equals the wave velocity. It is this added forward velocity that causes a greater water rise than predicted.

To compensate for this orbital velocity, the expression $\frac{h}{2}$ in Eq. (29) should be substituted by a new value H, defined by

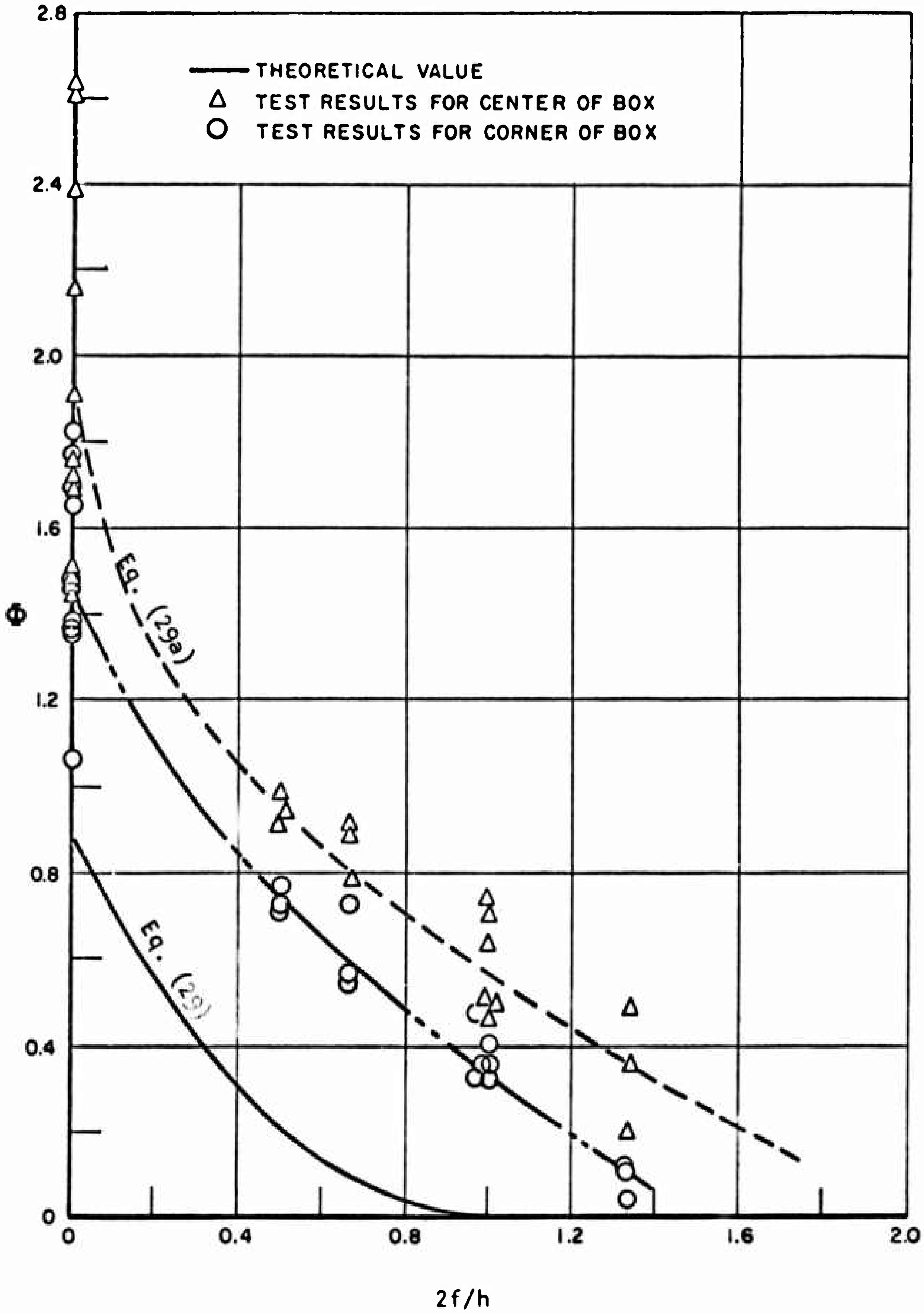


FIGURE 14. Comparison of Theoretical and Experimental Shipping-Rate Factor (Φ) with Freeboard-to-Wave-Height Ratio ($\frac{2f}{h}$)

$$H = \frac{h}{2} + C \frac{v^2}{2g} \tag{30}$$

where v = wave velocity

C = an empirical coefficient

If, for these tests, a value of C equal to 1/3 is used, the measured data come near the predicted data (the dashed line in Fig. 14). Therefore, the following improved version of Eq. (29) should be employed (until a better one comes along):

$$Q_{ave} = \frac{3.33L}{\pi} \left(\frac{h}{2} + \frac{v^2}{6g}\right)^{3/2} \Phi \tag{29a}$$

Since v is a function of wave length (see Eq. [17]), it would be about 20 ft/sec for a period of approximately 4 seconds.

Equation (29a) can be used to determine trade-offs between freeboard and bilge pumping, for various expected wave conditions.

Damage to Hull Integrity

The amount of water which penetrates into a damaged vehicle-hull depends on the size of the hole in the hull and its location with respect to the waterline. For this calculation the following formula may be used:

$$Q = kA\sqrt{2gh}$$

where Q = amount of water shipped, cfs

A = sectional area of the hole, sq ft

h = distance from the hole to the waterline, ft

g = gravitational constant, ft/sec^2

k = a coefficient determined experimentally, usually near 0.8

A ship is made less sinkable by subdividing the whole watertight volume into isolated compartments with transverse and longitudinal bulkheads, a double bottom, and watertight decks and platforms. The number of transverse watertight bulkheads depends upon the kind of ship, the kind of cargo carried, and the water-plane area. Amphibious vehicles should be secured against sinking by the use of structural and technical arrangements (installation of bailing pumps, sealing of the hull, etc.) However, structural and technical arrangements cannot provide complete unsinkability unless they are skillfully employed. A well-instructed and trained crew is essential for all operations.

Chapter 2

WATER SPEED

The concepts of thrust and drag are the fundamental working parameters of hydrodynamics and naval architecture. Water speed may be increased by either of two obvious methods: an increase in thrust, or a reduction in drag. An increase in thrust, however, involves a corresponding increase in installed power and expended fuel, and a reduction in performance efficiency. For this reason, great efforts have been expended in the study of vehicle hydrodynamic drag. The following section will discuss the drag of bodies with land-vehicle shapes. Propulsion will be discussed immediately thereafter.

THEORY OF HYDRODYNAMIC RESISTANCE

The precise drag for an object immersed in a fluid can be predicted only if the complete pressure and velocity distribution around the body can be accurately determined. The appropriate fluid dynamics equation, defining the relationship between pressure, velocity, viscosity, etc., is the well-known Navier-Stokes equation.[4] Solutions of the Navier-Stokes equation, however, exist for only a few restricted limiting cases of laminar flow, where either dynamic effects or viscous effects are negligible. If the viscous and dynamic effects are both present, as is usual in the case of a vehicle moving in water, the Navier-Stokes equation is non-linear and virtually unsolvable, analytically. Therefore, solutions by high-speed digital computer are being investigated, much as solutions of complex non-linear heat-transfer problems have been employed in the aircraft engine field. This attack, however, is, for practical considerations, suitable only as a design guide, rather than as an evaluation technique. The solutions so far developed have been useful only for the simplest shapes and are, at present, accurate only to about 25 percent.

Naval architects continue, therefore, to rely heavily on scale-model tests for prediction of the resistance inherent in a proposed design. Theories developed by Buckingham, Froude, Schoenherr, and others in the

field of similitude,[14] have been used to test and evaluate a vast number of model shapes and sizes. Coupled with some fundamental theories associated with scale-model tests, a reasonable prediction of the performance of standard vehicle shapes can be made without testing, and constructive changes to improve performance can be suggested.

Although the total drag force acting on a vehicle immersed in a fluid is a complex interrelationship of many factors, many researchers in the field like to consider the total drag as roughly divisible into three main elements: skin friction, form (pressure) drag, and wave resistance.

Skin Friction

Skin friction is considered that portion of the total drag resulting from the viscous fluid forces immediately adjacent to and acting on the surface of the vehicle. This drag may be thought of as the same kind of drag experienced by a fluid flowing in a pipe. Experience has shown that the skin friction drag is, in general, approximately proportional to the wetted surface area and to the 1.8 power of speed:

$$D_f = KA_{sw}v^{1.8} \tag{30}$$

where D_f = drag force due to skin friction

A_{sw} = submerged wetted area

v = speed, ft/sec

K = constant of proportionality

Although no good theoretical treatment of skin drag exists, Schoenherr, in a series of famous experiments, was able to separate the skin friction drag and develop the following empirical equations[3] which can be used to predict the skin friction on a partially submerged body:

$$\frac{0.242}{C_f} = \log_{10}\left(\frac{v\ell}{\nu} C_f\right) = \log_{10}(R_n C_f) \tag{31}$$

$$D_f = (C_f + C_k)\, \rho \frac{v^2}{2} A_{sw} \tag{32}$$

where C_f = friction coefficient (a function of Reynolds number)

ν = fluid kinematic viscosity

R_n = Reynold's number, $\frac{v\ell}{\nu}$

C_k = Kempf roughness coefficient (a function of the surface roughness)

ρ = fluid density

Typical values of C_f and C_k, for fresh water at 59°F, are:

R_n ($\times 10^6$)	C_f ($\times 10^{-3}$)	C_k
1.5	4.083	0.0004
2.0	3.878	0.0004
2.5	3.719	0.0004
3.0	3.600	0.0004

As indicated in Eq. (32), the skin friction drag on ships is primarily a function of the surface roughness and wetted area. In the case of an amphibious land vehicle, however, the surface roughness is grossly overshadowed by the turbulence caused by vehicle appendages such as axles, drive shafts, suspensions, and wheels. Figure 15 shows the results of modifications to model hulls of the amphibious DUKW[15] and LARC.[16] The DUKW model had its wheels and axles shrouded by streamlined covers. The LARC model had its wheels and axles removed and the wheel wells filled in. Thus Fig. 15 shows the drag contribution of these appendages. The "Hay block" in the figure is a rectilinear box of similar dimensions and

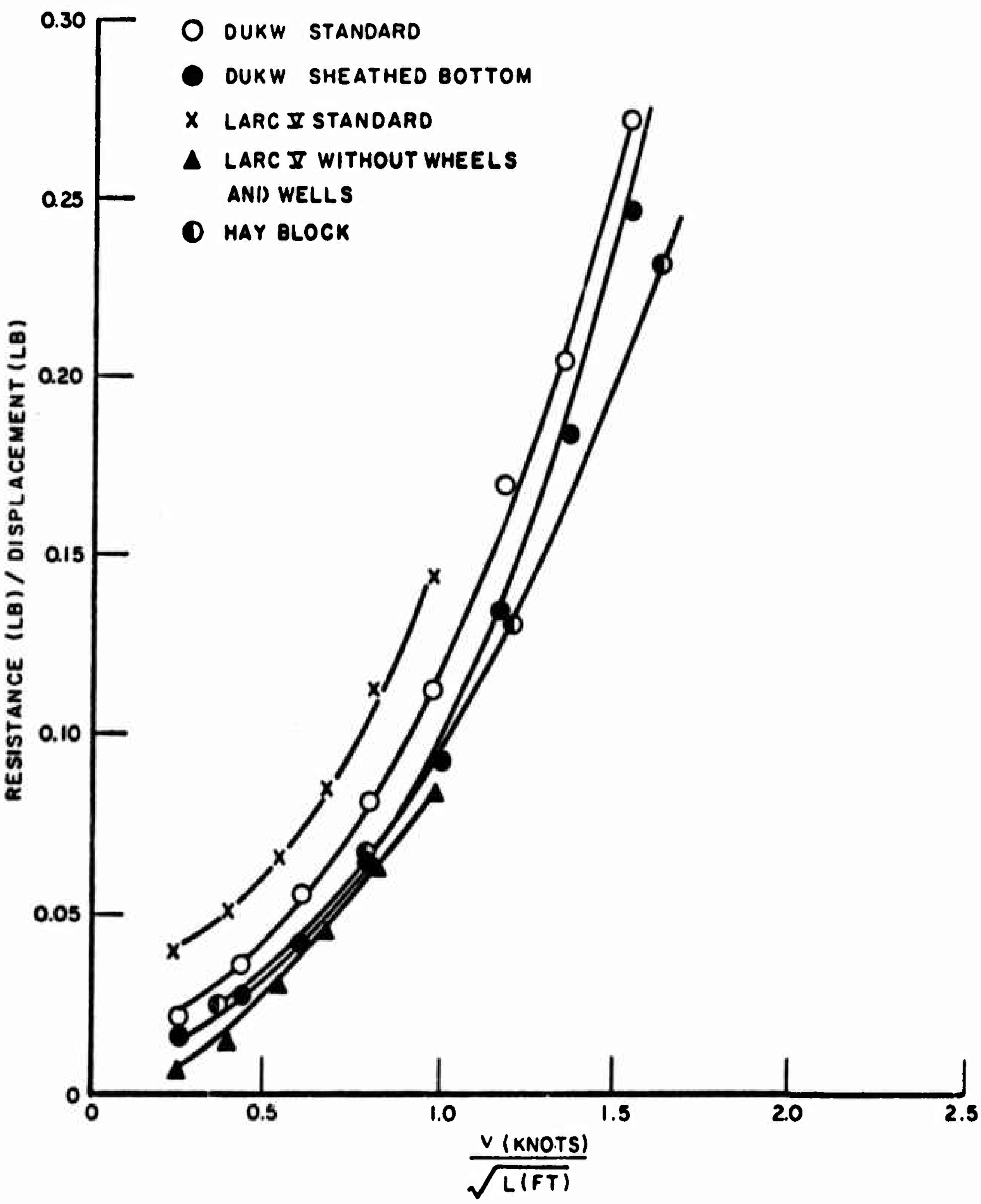


FIGURE 15. Effect of Appendages

displacement for comparison purposes.

Figure 16 shows the differing results obtained for a model of the DUKW when a sheath of sheet metal is used to cover the many appendages and smooth out the underbody. Vehicle designers should therefore take special care to round all vehicle edges and to sheath vehicle appendages wherever possible.

Form Drag

The form (or pressure) drag is the resultant of all pressure forces acting about the body of the vehicle. It is a combination of the positive static pressures on the front of the vehicle and the negative pressures about the rear. In numerous tests, it has become evident that the form drag is also approximately proportional to the 1.8 power of the speed and roughly proportional to the width of the vehicle's wake. This statement does not imply that the bow has no influence on form drag; it does. However, greater benefits will usually derive by proper treatment of the stern.

Wave Resistance

The force of wave drag is the result of the energy expended in creating waves as the vehicle pushes through the water. Major waves are generated at the bow and stern of the vehicle and at all points where there are abrupt changes in vehicle shape. Waves are a lifting of the water against gravity, and it has been established that wave drag is related closely to the dimensionless number relating gravity forces to inertial forces. This number is called the Froude number:

$$N_{Fr} = \frac{v}{\sqrt{g\ell}} \tag{33}$$

where g = gravitational constant

ℓ = a characteristic length

In some cases, the characteristic length is taken as the square root of a characteristic area or the cube root of a characteristic volume.

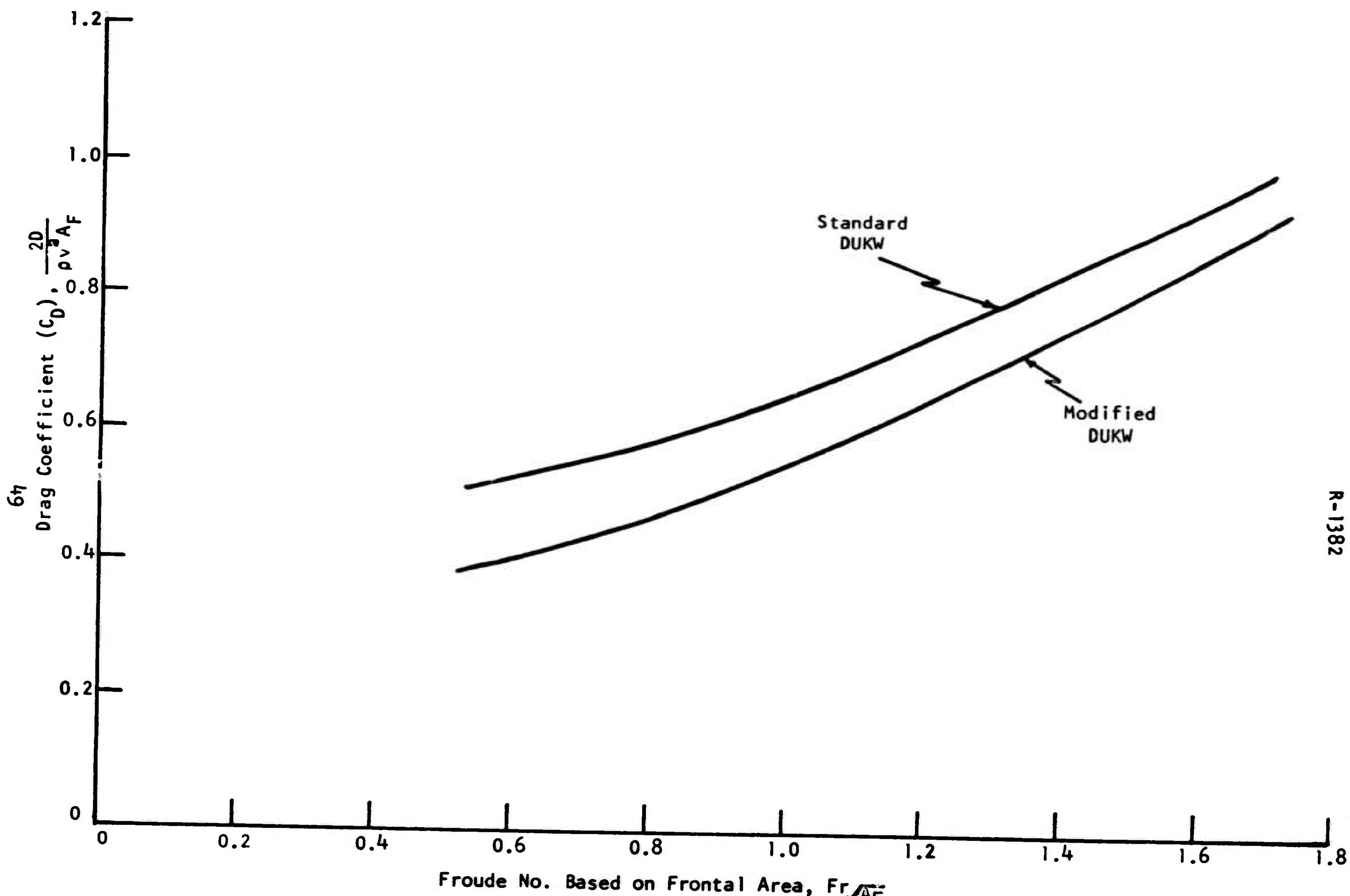


FIGURE 16. Comparison of Standard DUKW with Sheathed-Bottom DUKW

Frequently, for the sake of simplicity, the term "drag coefficient," C_D, is used to represent the sum of all drag forces. As such, the total drag may be computed by

$$D = \frac{1}{2} C_D \rho A_F{}^2 \tag{34}$$

where A_F = the submerged frontal area

C_D = drag coefficient

Here C_D is usually not constant and is presented as a function of velocity, Reynolds number, or Froude number, as in Fig. 16.

Many tests have demonstrated that the wave drag in displacement-hull vessels is roughly proportional to the 4th power of the speed of the vehicle. An examination of the curves resulting from model and full-scale drag tests[17] (Fig. 17) readily shows how drag increases at a relatively gradual rate at low speeds, but rapidly accelerates until there is almost a "wall" of resistance. This "wall" is a direct result of that 4th power relationship. Naval architects are able to predict, quite accurately, the speed at which this "wall" occurs, by using a simplification of the Froude number called the "speed/length ratio."

$$\text{Speed/length ratio} = \frac{v}{\sqrt{L}} \tag{35}$$

where v = speed, knots

L = length of vessel, ft

The wave-resistance "wall" occurs at approximately the speed at which the speed/length ratio equals 1 (again see Fig. 17).

Briefly, let us consider the example of a jeep-size (approximately 10-ft long) vehicle design. The speed that corresponds to a speed/length ratio of 1 for this vehicle is $\sqrt{10}$ = 3.16 knots or about 3.5 mph. For a 1000-ft ship, however, the speed/length ratio approaches 1 at near 30 knots. It then becomes apparent that no jeep-size vehicle is going to achieve a

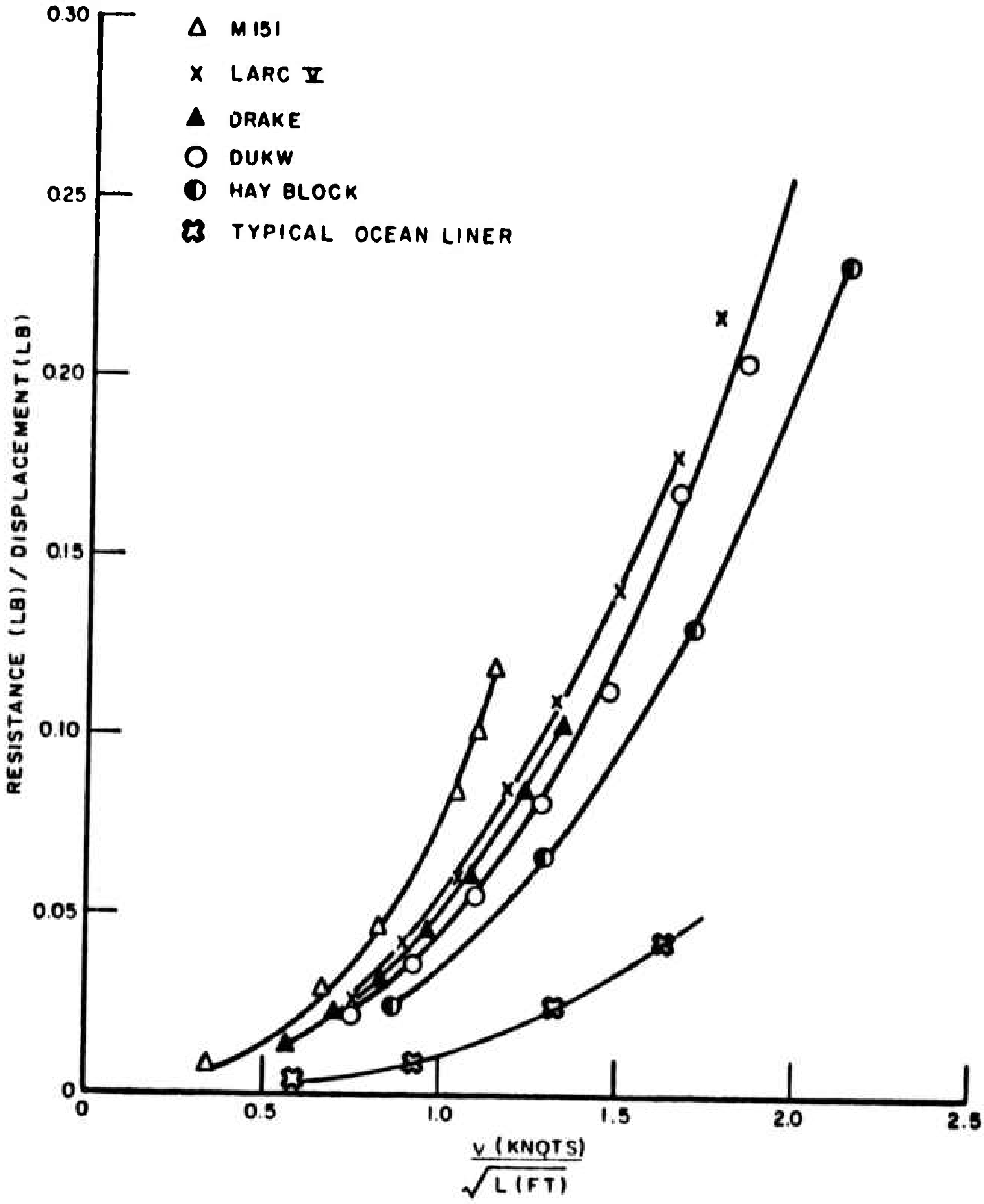


FIGURE 17. Specific Resistance of Displacement Amphibians vs. Speed/Length Ratio

water speed much greater than 3.5 mph unless some radial change in hull form, horsepower, or propulsion is made, or unless flotation of other than the displacement type (hydrofoil, planing hull, etc.) is employed.

Naval architects have long since proven that it can become exceedingly expensive, as regards both power and operating cost, to operate a displacement-type vessel much beyond a speed/length ratio of 1. It behooves the designer of a displacement-type amphibian to consider this point as a good indication of the maximum speed he may obtain without undue expense. However, further increases in hull length, such as might do any good, are usually prohibited for amphibians, because of land utility. Thus the water speed of a reasonably sized contemporary amphibian reaches a limit of 6 to 8 mph.

Consequently, other approaches must be taken if further increases in water speed are to be achieved. One such approach might be to connect displacement-type vehicles, to form a train. A configuration of this kind could operate in the water as a long, slender, vehicle; hence at a low speed/length ratio. On land this same vehicle could operate in the train configuration or could be disconnected, with units operating independently. An articulated train made up of, say, 10 vehicles, each 30-ft long, offers the theoretical capability of approximately 17 knots at $\frac{V}{\sqrt{L}} = 1$. Model tests[11, 18] have shown (Fig. 18) that the specific resistance of vehicle trains drastically decreases with increase in train length. Other model tests[19] have shown that five LVTP-5 vehicles connected in tandem can operate at a speed 50-percent greater than the maximum of an individual vehicle, if all vehicles are generating maximum thrust.

PLANING HULLS

One approach to the achievement of greater water speed is the use of the dynamic lift created under the hull to support the load. Many different hull shapes, with their respective lift/drag ratios and their rough-sea characteristics, are well known and documented. However, an amphibious vehicle whose required load and size are commensurate with those of a comparable land vehicle is grossly overloaded by conventional planing-boat

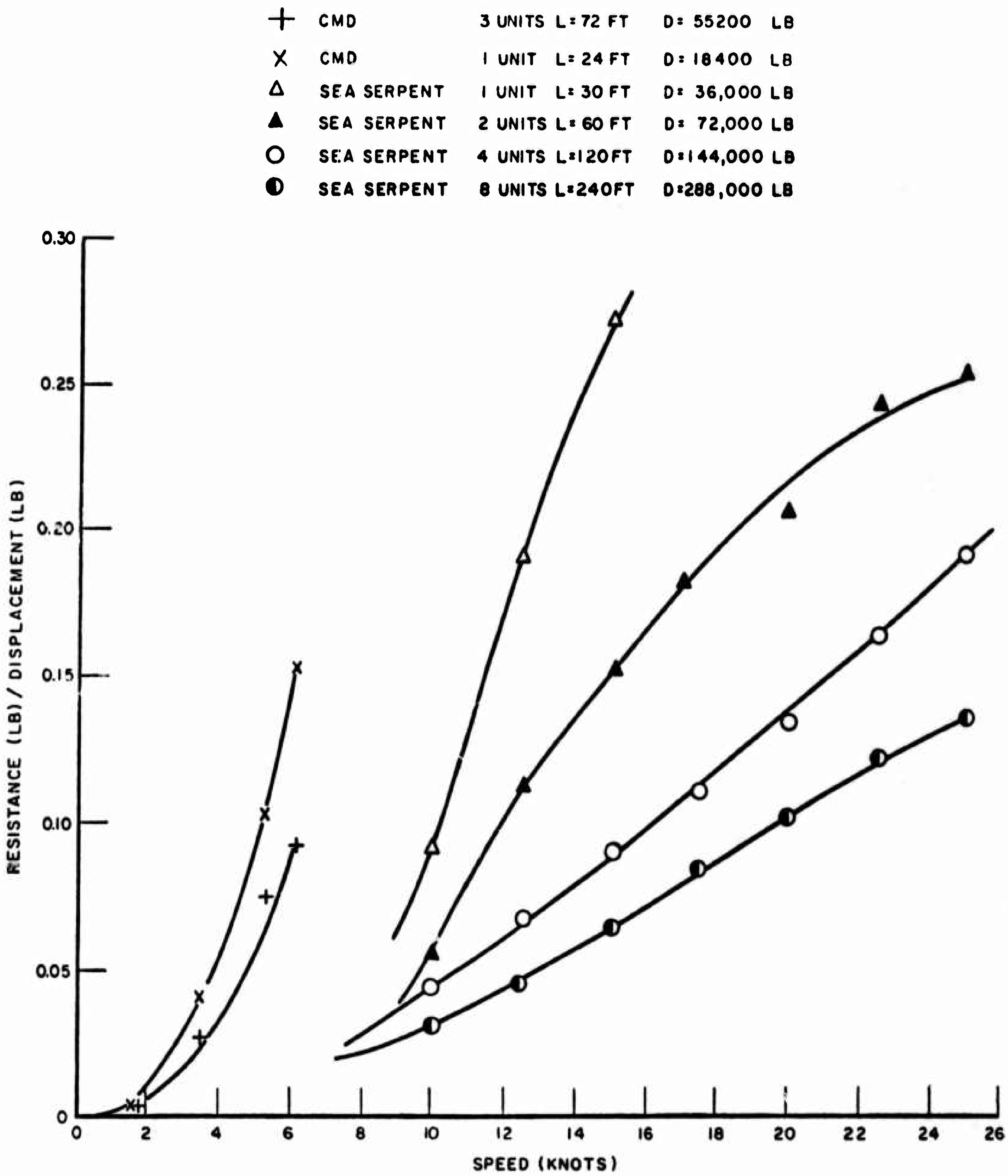


FIGURE 18. Specific Resistance of Coupled Amphibians[11,18]

standards. A 5-ton payload amphibian 40-ft long with a 10-ft beam would weigh a minimum of 35,000 lb. A pure planing hull of such dimensions should not weigh much above 20,000 lb. Bottom loadings of a good high-speed planing hull rarely exceed 70 lb per square foot, whereas for an amphibian it is difficult to attain a bottom loading even as low as 120 lb per square foot. Similarly, the lift/drag ratio of a planing-type amphibian cannot begin to approach that of conventional craft. It is therefore doubtful whether a true planing hull can be achieved for an amphibian. Partially planing hulls, however, have been built and operated (the LVW).

Complicating the picture still further is the fact, established in the laboratory,[20] that wheel well closures (such as are used in aircraft) can receive up to 6g impact loads in a high-speed operation in moderate waves. Such impacts can render any reasonable door structure and mechanism useless. Thus the wheel wells must remain open. Although wheel-door and operating-mechanism weight can then be saved, careful consideration must be given to the flow pattern past the wheel openings, so that flow disturbance and loss of lifting surface can be kept at a minimum. Wheels must be clear of any spray, yet must be retractable with a minimum of mechanism and at the same time capable of suspending and supporting the load of the vehicle and its cargo. Of no help is the fact that a vehicle of such weight needs tires of considerable size for reasonable land mobility.

The resistance characteristics of two amphibious planing hulls, one with open wheel wells and one with closed wells,[20] and each designed to carry 5 tons of cargo at 25 mph, are shown in Fig. 19. The resistance characteristics of a standard type of displacement hull without wheels[21] but of comparable weight and size, and a composite of seventeen conventional planing hulls,[22] are included as yardsticks.

Figure 19 shows how the resistance curves follow the conventional displacement-hull pattern until the dynamic lift on the hull becomes effective and the hull begins to plane.

For a planing hull, however, the reduced drag at higher speeds can be used to advantage only in relatively calm water. In rough seas, the speed capability of the planing hull is drastically reduced, because of two major factors: the power requirement is much greater (as much as 60 percent

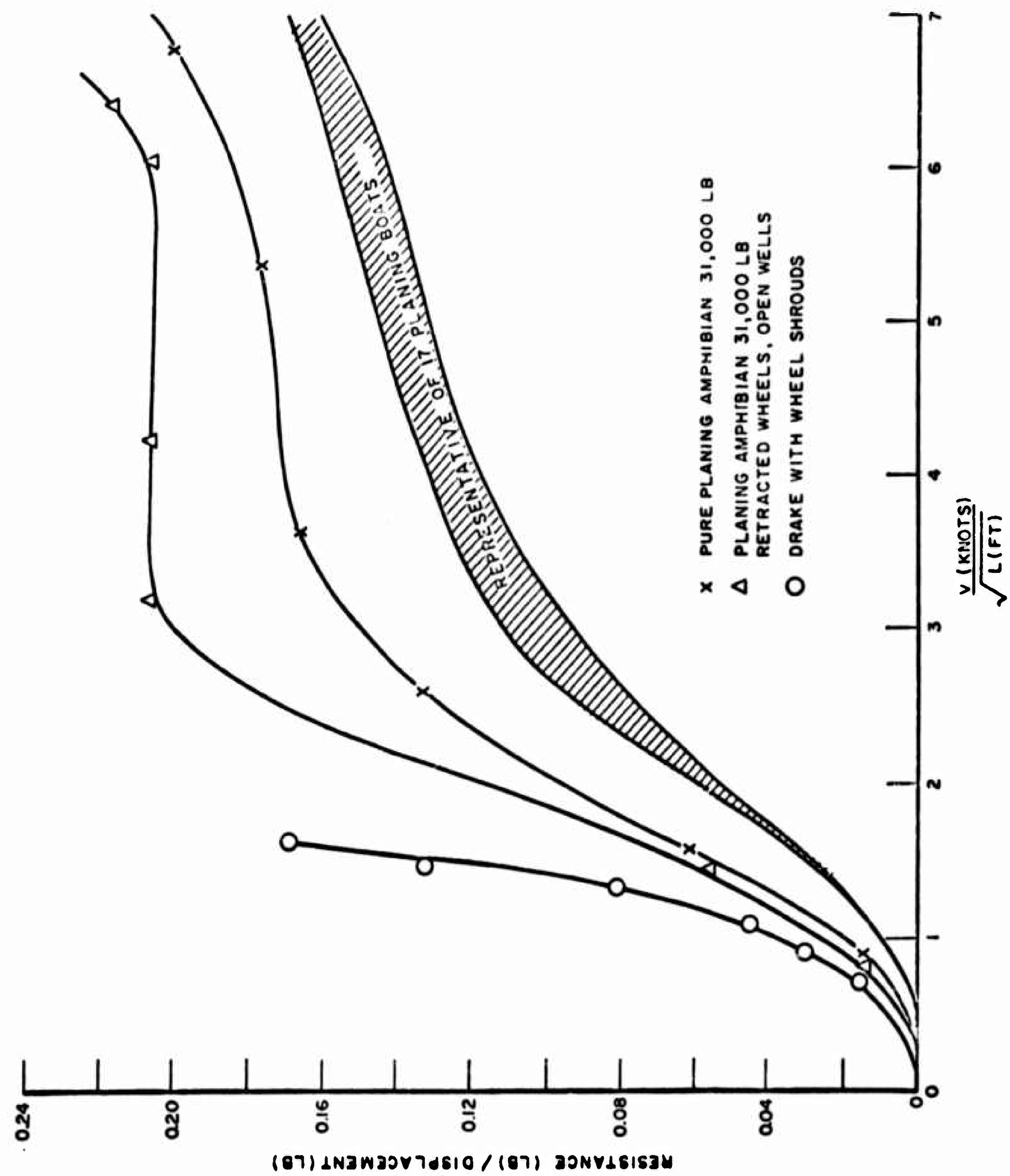


FIGURE 19. Specific Resistance of Planing Hulls

in 3-ft waves),[20,23] and the vertical accelerations exceed human tolerance limits. Therefore, the top speed in a 3-ft head sea may well be below 15 knots.

HYDROFOILS

Hydrofoils are direct parallels of airfoils, except that they operate submerged in water. Their dynamic coefficients are unaffected by type of fluid, but the lift generated by a foil is in proportion to the fluid density. A foil in water will therefore generate about 800 times the lift it would in air. However, the amphibian's speed is much below that of the airplane.

To minimize installed horsepower, the hydrofoil should generate a lift of at least 50 percent of vehicle weight at 6 knots, in order to lift the hull partially out of the water and reduce the displacement drag. Full vehicle lift should be generated by the time speed reaches 12 to 15 knots. The lift of a hydrofoil increases roughly as the square of its size, but the weight of the vehicle increases as the cube; therefore a small vehicle has a better chance for hydrofoil support. Thus a vehicle double the size of the small vehicle, with a foil of the same proportions, will weigh eight times as much as, but generate only four times the lift of, the smaller vehicle. And the larger the vehicle, the more difficult it is to stow the hydrofoils for land travel.

Hydrofoils can be of either the surface-piercing or the fully submerged type. Surface-piercing types are self-equalizing in lift and roll, since the lifting surface (and hence the lifting force) varies directly with the depth of submergence. A disadvantage of surface-piercing foils, however, is their tendency to rise and fall with the wave pattern, especially with long waves. Riding in a vehicle equipped with these foils can therefore be uncomfortable. And the foil, since it is near the surface, may -- if the vehicle is traveling at or near the same speed as the waves and in the same direction (i.e., in following seas) -- take on an attack angle different from that intended and lose lift, because of the orbital motion of the water. Under such conditions, this kind of vehicle must be

operated with care.

A fully submerged foil will eliminate these shortcomings, but the vehicle will become more complex because such a foil is not self-equalizing, and an autopilot type of control system is required. Roll angle and elevation above the water must be sensed, damped, averaged, and converted into specific input to the foil's control surfaces. The stability of the vehicle in pitch and roll depends entirely on this system. The vehicle is therefore flown practically like an aircraft, though constantly within the small margin-of-error region associated with aircraft take-off and landing. The advantage of hydrofoils at the higher speeds, so far as hydrodynamic resistance is concerned, shows clearly in Fig. 20. But if take-off power requirements are to be kept within reason,[24,25] the amphibian's road-wheels must be retractable. Also, an exceedingly responsive propulsion system -- such as a variable pitch propeller -- is mandatory so that the vehicle can operate over the complete speed and torque range required of it.

AIR-CUSHION VEHICLES

The air-cushion vehicle represents an attempt to lift the entire hull out of the water to eliminate hydrodynamic drag entirely. This type of vehicle relies on the generation of a small positive pressure underneath the vehicle to keep it above the surface. In general, the lift forces are increased with an increase in the area under the vehicle, but are greatly decreased with an increase in height above the surface. A large vehicle of this kind generally benefits from its size; it can lift higher above the surface and do so more economically. Power is a function of the area of the exposed perimeter; load-carrying capability is a function of plan-form area. Doubling the plan-form area and the cargo space require only a 40-percent increase in power. Unskirted vehicles --

(1) Are inefficient in the considered size.

(2) Have poor directional stability.

(3) Are incapable of climbing the slopes so commonly encountered in beaches and other water-to-land transportation zones.

(4) Are also quite unstable when operated in waves.

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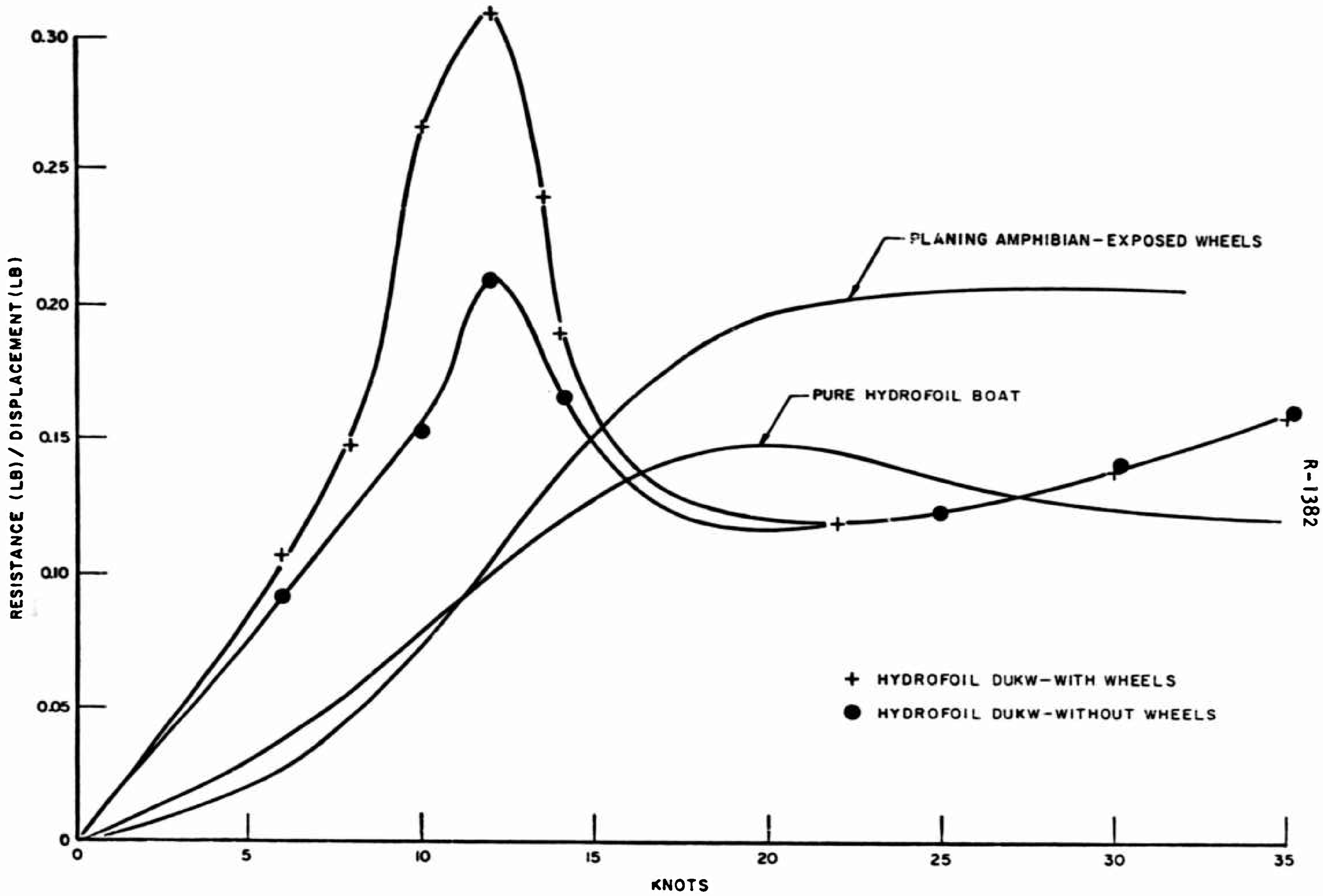


FIGURE 20. Specific Resistance of High-Speed Amphibians

Therefore only the skirted air-cushion vehicles are presently being considered. In this configuration, the weighted skirts suspended from the periphery of the vehicle help to keep the pressurized air beneath the vehicle. They have materially improved ACV operations in waves and over land obstacles.

The skirted vehicles have exhibited excellent performance characteristics over fairly rough seas, and over irregular terrain. They have been extremely useful in Vietnam on riverine operations. They have also been quite successfully used in the commerical sector (in ferrying). Major operational problems, to date, are the excessive noise and dust generated by the vehicle.

PREDICTIONS OF VEHICLE DRAG

Since the art of making theoretical predictions of vehicle drag is imperfect, the designer must, to predict the drag of a known shape, perform a series of model tests. The measured drag on the model is the sum of the friction, form, and wave components. Reference to scaling laws[4,14] indicates that the friction drag increases in direct linear proportion to the scale, while form and wave drag increase as the cube of the scale.

From Eqs. (31) and (32), the skin friction of the model can be calculated from underwater photographs showing the wetted area. It can then be subtracted from the measured model drag, and the remaining drag multiplied by the cube of the scale factor, to obtain the full-scale form and wave drag. Full-scale frictional drag is then calculated from Eqs. (31) and (32) and added to the predicted form and wave drag to obtain the total predicted full-scale drag force. In practice, the skin drag of small ships (such as the amphibious vehicles under consideration) is usually small as compared with form and wave resistances; often, direct scaling by the cube factor is employed with little error.

COMMON SHAPES

This section is intended to serve as a guide for the designer of military amphibious vehicles in estimating the drag of various conventionally shaped bodies within a reasonable engineering accuracy, without recourse to model tests.

Reports on the testing of specific vehicles appear throughout the literature. A number of the reports which relate to amphibious vehicles have been collected,[15,16,21,28] and data taken from them are presented here in Figs. 17, 21, 22, and 23. Plots are presented with different, frequently used, coordinates. Figure 17 is a comparison of specific resistance (the drag force divided by the weight of water displaced) with the speed/length ratio. Figure 21 is a presentation of the drag coefficient (C_D) for various amphibians, as a function of Fr_L (Froude number, based on over-all length). Figure 22 is a presentation of the same data as a function of $Fr_{\sqrt{A}}$ (Froude number, based on the square root of the submerged frontal area). And Fig. 23 is a comparison of specific resistance, $\frac{R}{\Delta}$, versus Fr_L. It may be noted that craft such as motor boats, planing boats, and hydrofoils cannot appear on these graphs; they operate at Froude numbers between 1 and 4.

Several efforts were made to develop a method for making rough predictions of amphibious-vehicle drag without recourse to model tests. Figure 22 indicates that the accumulated data group themselves into the following three distinct bands:

<u>Band A</u>. Vehicles whose design represents considerable effort to achieve boat-like shapes (LARC V and LVTP-5). These vehicles form, as expected, a lower-limit band of resistance starting with a C_D of approximately 0.4 at low speeds and increasing to near 0.6 at $Fr_{\sqrt{A}} = 1.4$.

<u>Band B</u>. Vehicles which are basically trucks, but which have had some sheet-metal skin added to hulls, for reasonable smoothness (DUKW, World War II Amphibious Jeep, SUPERDUCK, and DRAKE). The drag coefficients of these vehicles start between 0.5 and 0.65 at low speed and increase to near 0.9 at $Fr_{\sqrt{A}} = 1.4$. Notable is the fact that a rectangular wooden block with a curved forefoot (front bottom edge) yields a similar drag

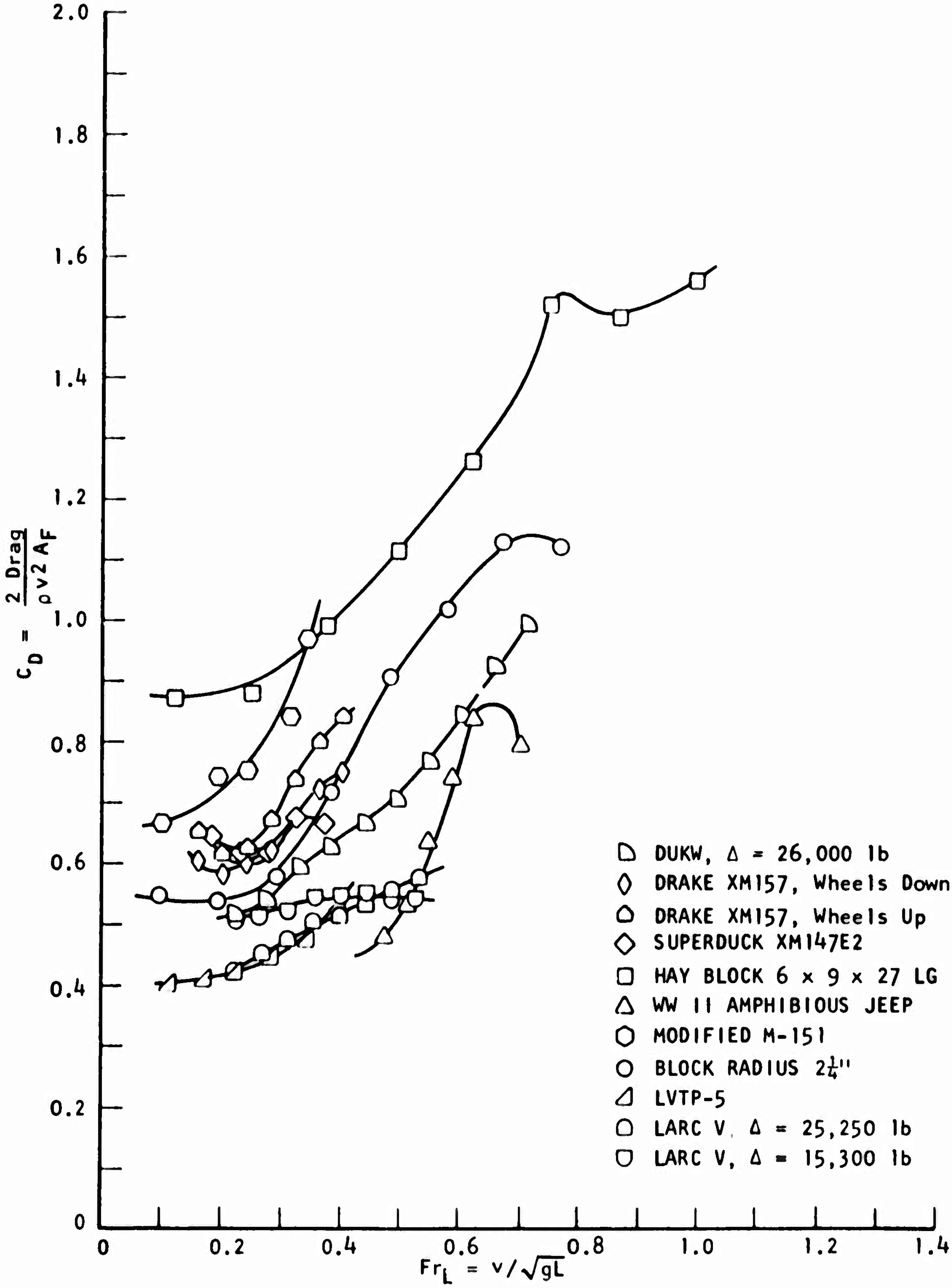


FIGURE 21. Drag Coefficient as a Function of Froude Number Based on Over-all Length

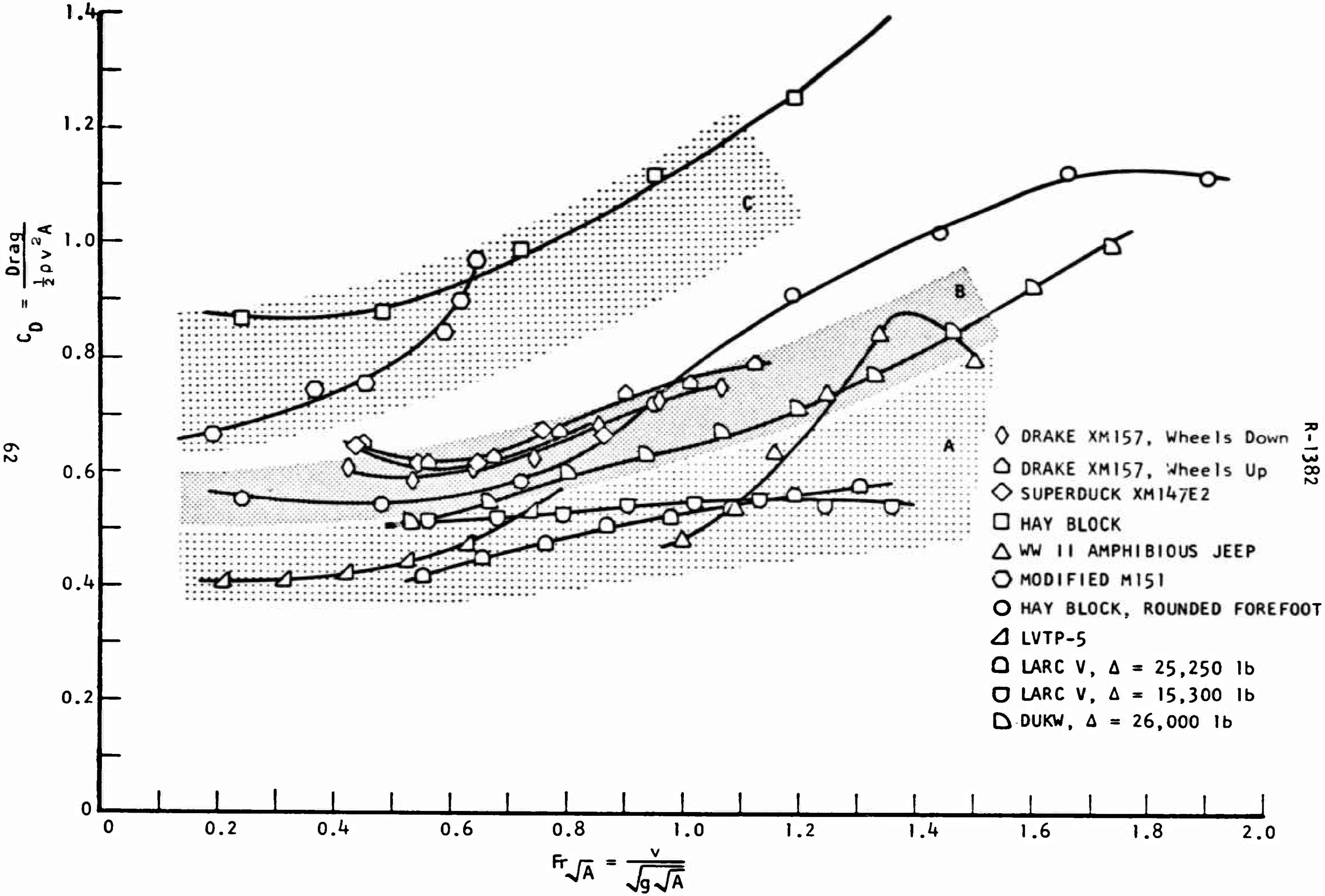


FIGURE 22. Drag Coefficient as a Function of Froude Number Based on Square Root of Submerged Frontal Area

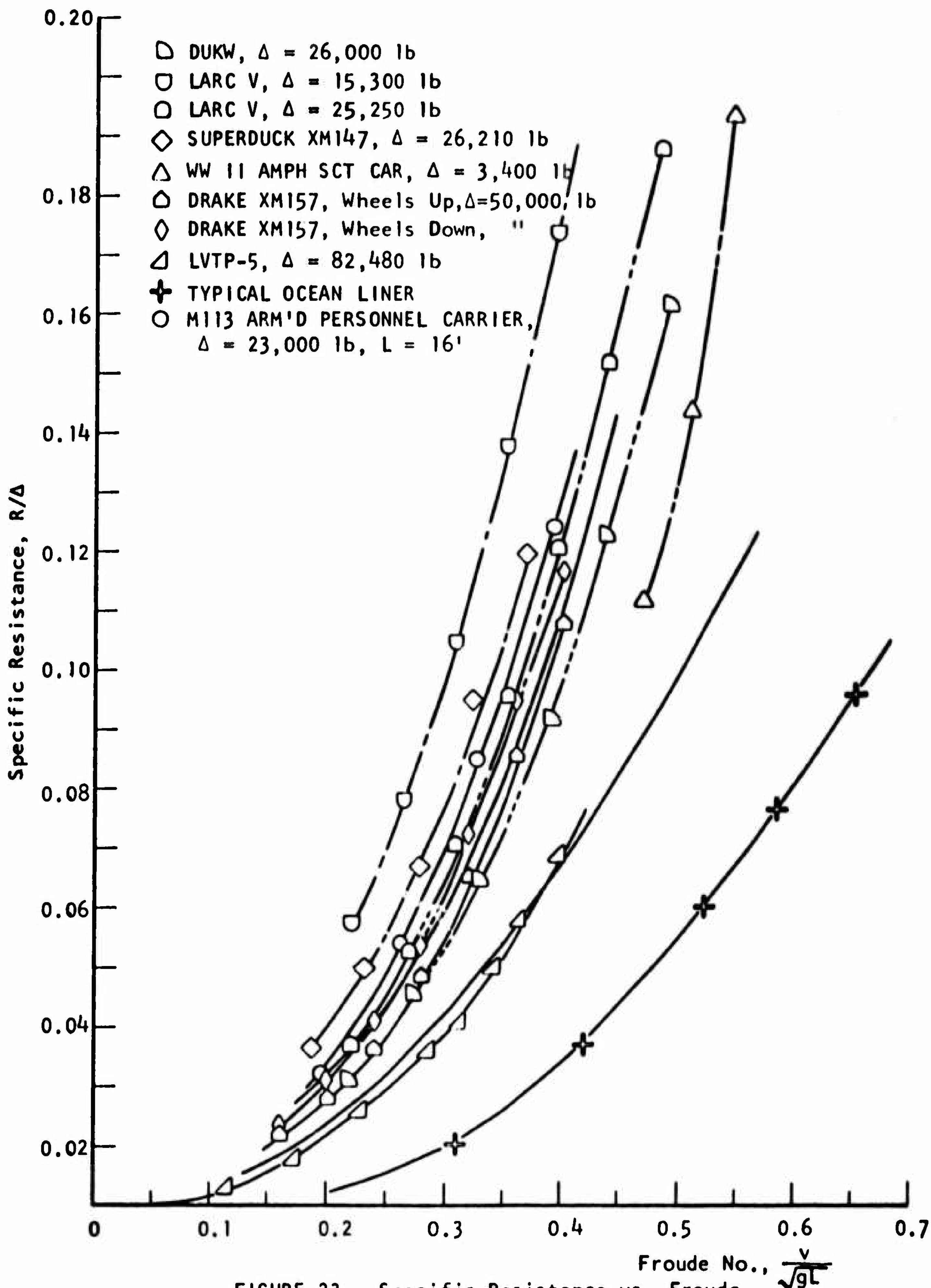


FIGURE 23. Specific Resistance vs. Froude Number Based on Over-all Length

characteristic over much of this range.

Band C. Land vehicles to which floating devices have been added without regard for hydrodynamic drag (Floating M151). The drag coefficient of this vehicle is near 0.65 at low speeds but quickly rises to above 1.0 at $Fr_{\sqrt{A}}$ = 0.6. Note that this curve is in the vicinity of a corresponding curve for a rectangular block, square on all edges.

Model tests are best for accurate determination of vehicle drag at various speeds. However, it may be possible to use Fig. 22 to estimate the expected drag of most conventional amphibians within ±20 percent of accuracy. First, the effort expended in making the hull boat-like would be considered, and the C_D estimated at various values of $Fr_{\sqrt{A}}$. The drag could then be calculated from Eq. (34).

Figure 23 is a plot of the data presented in Figs. 21 and 22, except that it shows how specific resistance varies with Froude number. It can be useful in estimating drag when the vehicle's length and weight are known, but not the submerged frontal area.

Experimental work done on many basic shapes can be said to divide into two notable categories. Hay[12,26] studied a number of barge-like hulls; Hoerner,[27] a number of simple ship forms. A description of work on the drag of many basic shapes, and, indeed, on the complete subject of ship resistance, is contained in Hoerner's book, Fluid Dynamic Drag,[27] and this book should be considered a prime reference for any study of hydrodynamic drag. Amassed therein is a body of information on the measured drag of many different shapes, and this information is valuable in making first-order approximations of vehicle drag. It is, in addition, an aid to the planning of design improvements. Works of this kind show the importance of considering certain body details.

One important characteristic is the shape of the forefoot of the vehicle. Figure 24 is a diagram of the tested models; Fig. 25 is a comparison of the drag coefficients of these models.

Also important are bow and stern treatments. Figure 26 (A, B, C, and D) shows the variety of shapes investigated by Hay.[26] The "ends" shown were attached to the same, simple, right parallelepiped body (indicated by

"Barge Body" in Fig. 26 with "End B" attached), either as a bow or a stern. The results of some of these tests are presented in Figs. 27, 28, 29, 30 and 31. The upper figure in each illustration shows plots of the specified end attached as the bow of the simple model; the lower, plots of the same end attached as the stern.

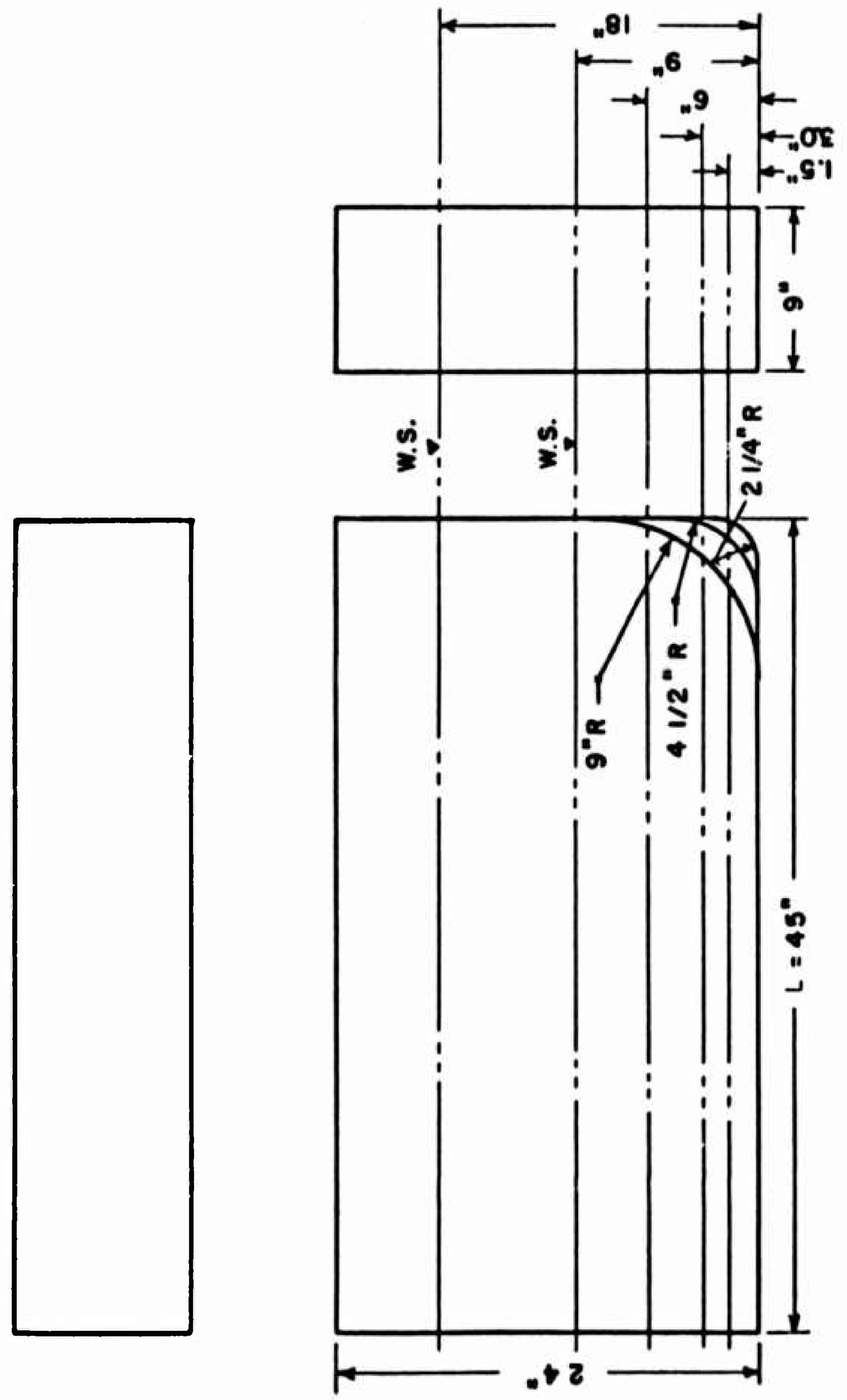


FIGURE 24. Rectangular Shapes with Varying Forefoot[12]

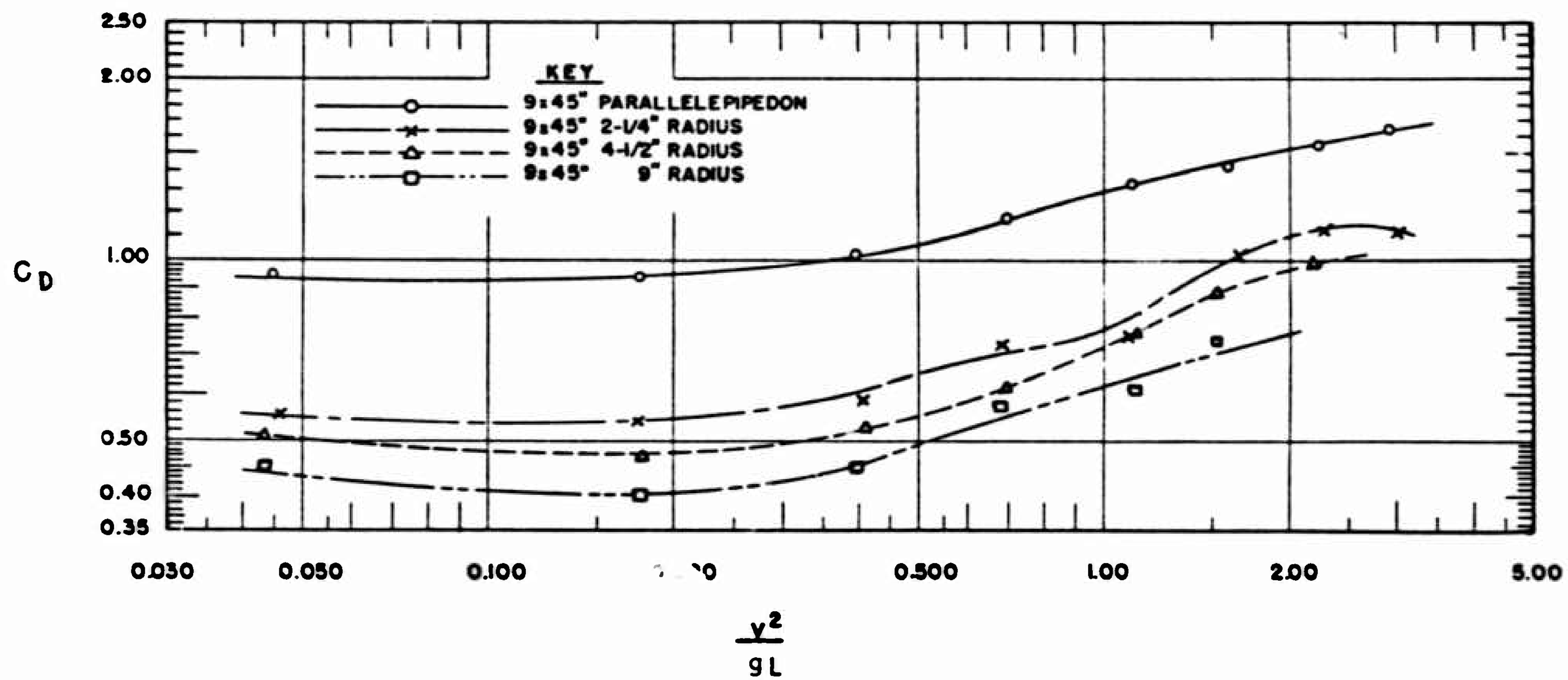


FIGURE 25. Effect of Forefoot Radius on Drag Coefficient[12]

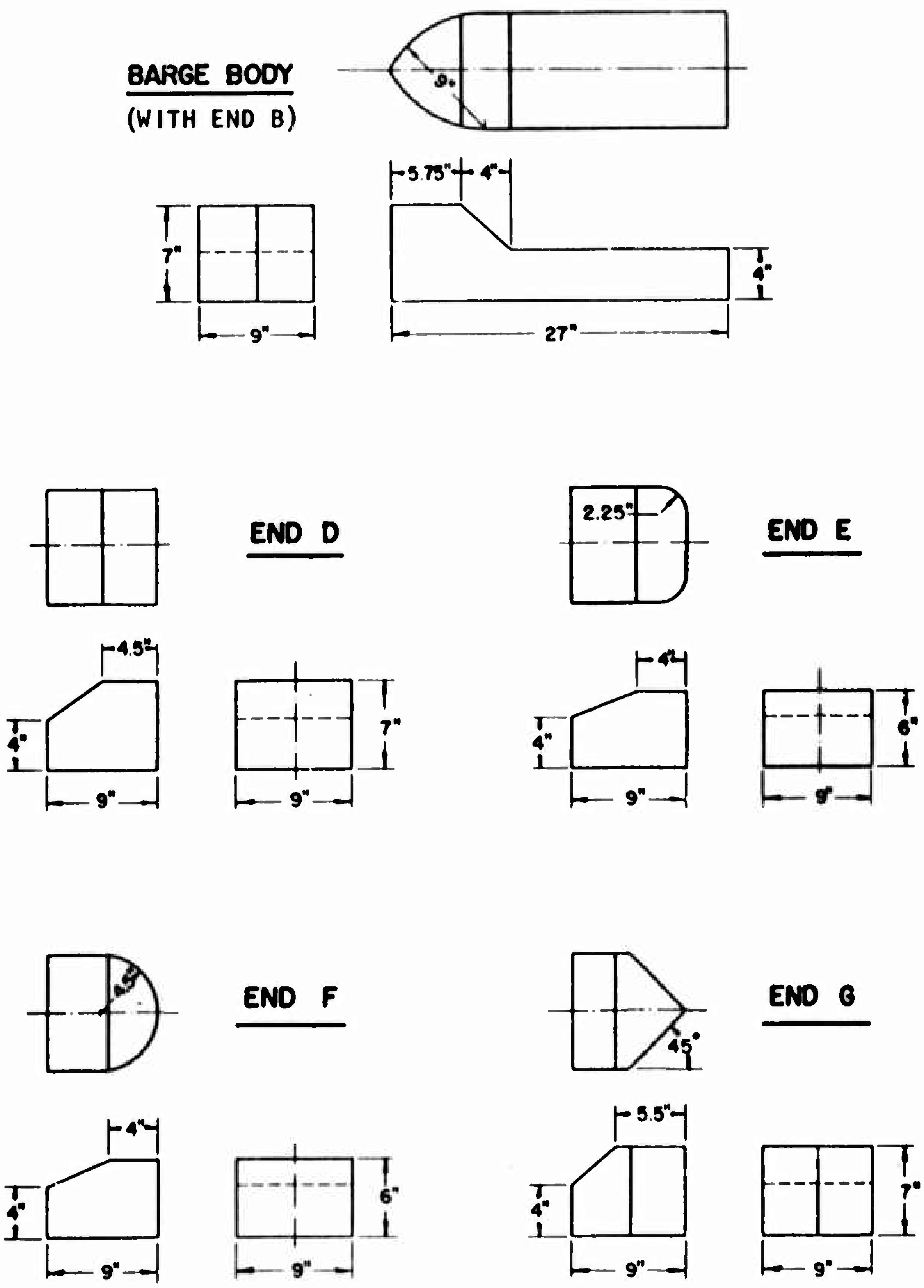


FIGURE 26A. Basic Bow and Stern Shapes[26]

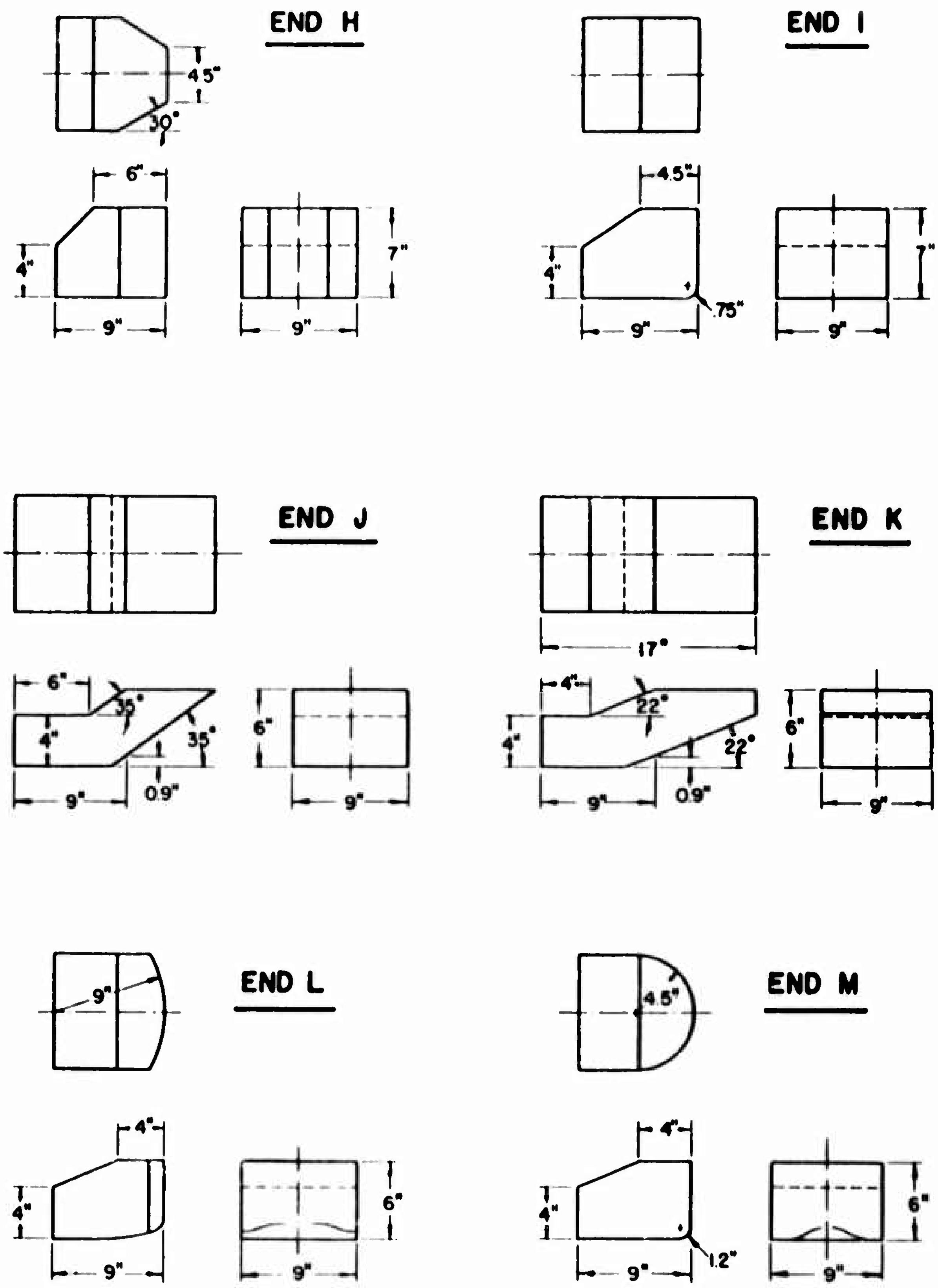


FIGURE 26B. Basic Bow and Stern Shapes[26]

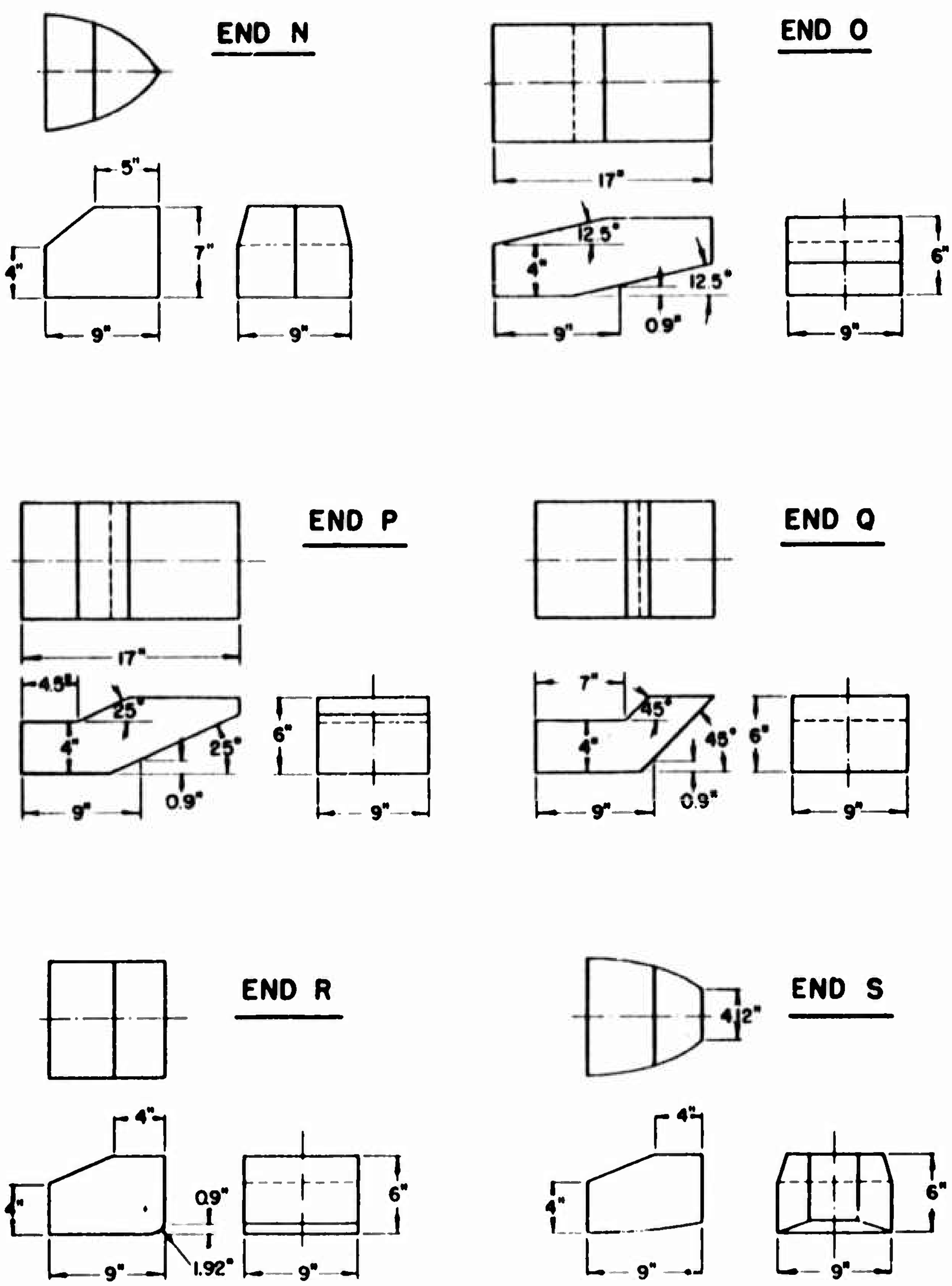


FIGURE 26C. Basic Bow and Stern Shapes[26]

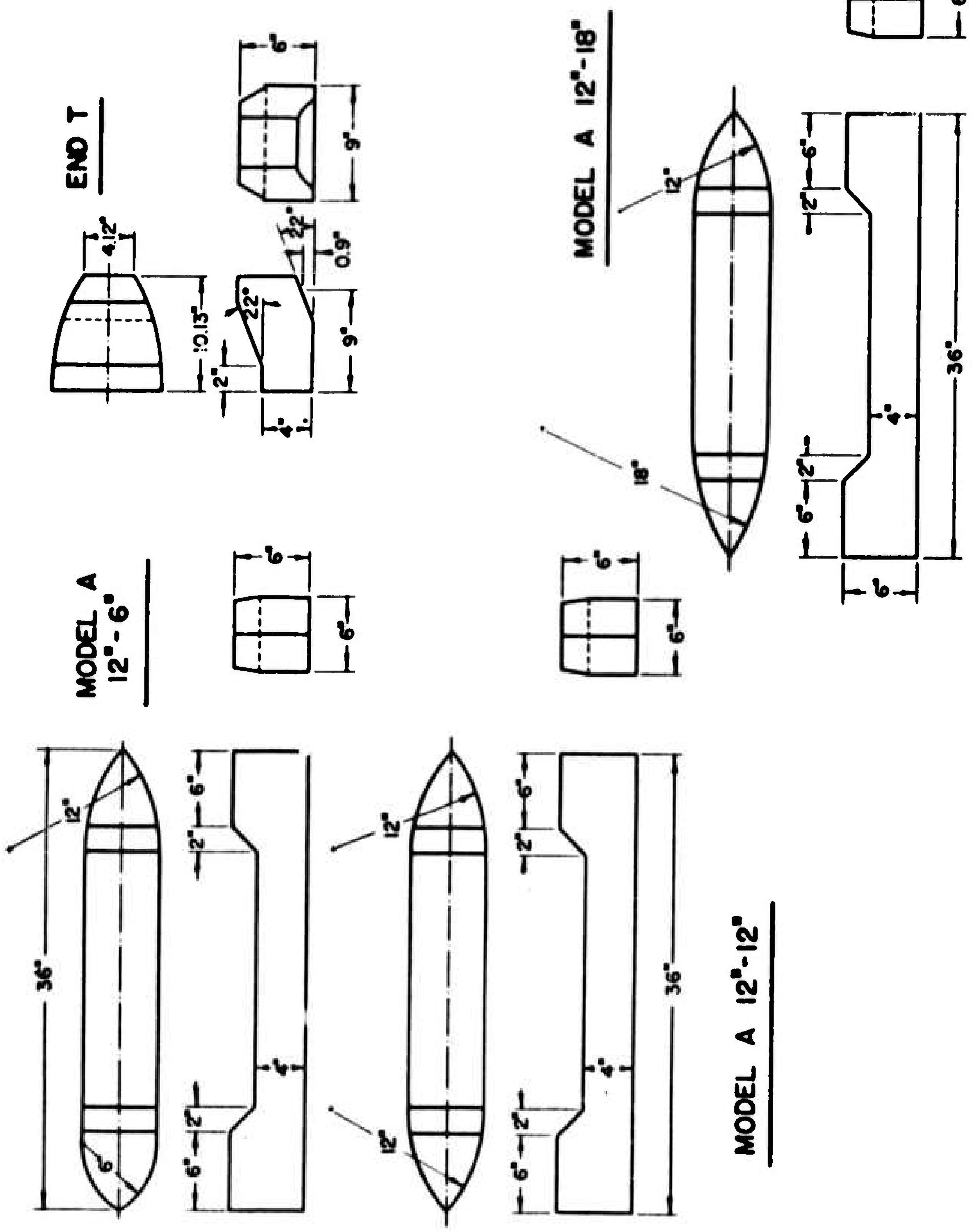


FIGURE 26D. Basic Bow and Stern Shapes[26]

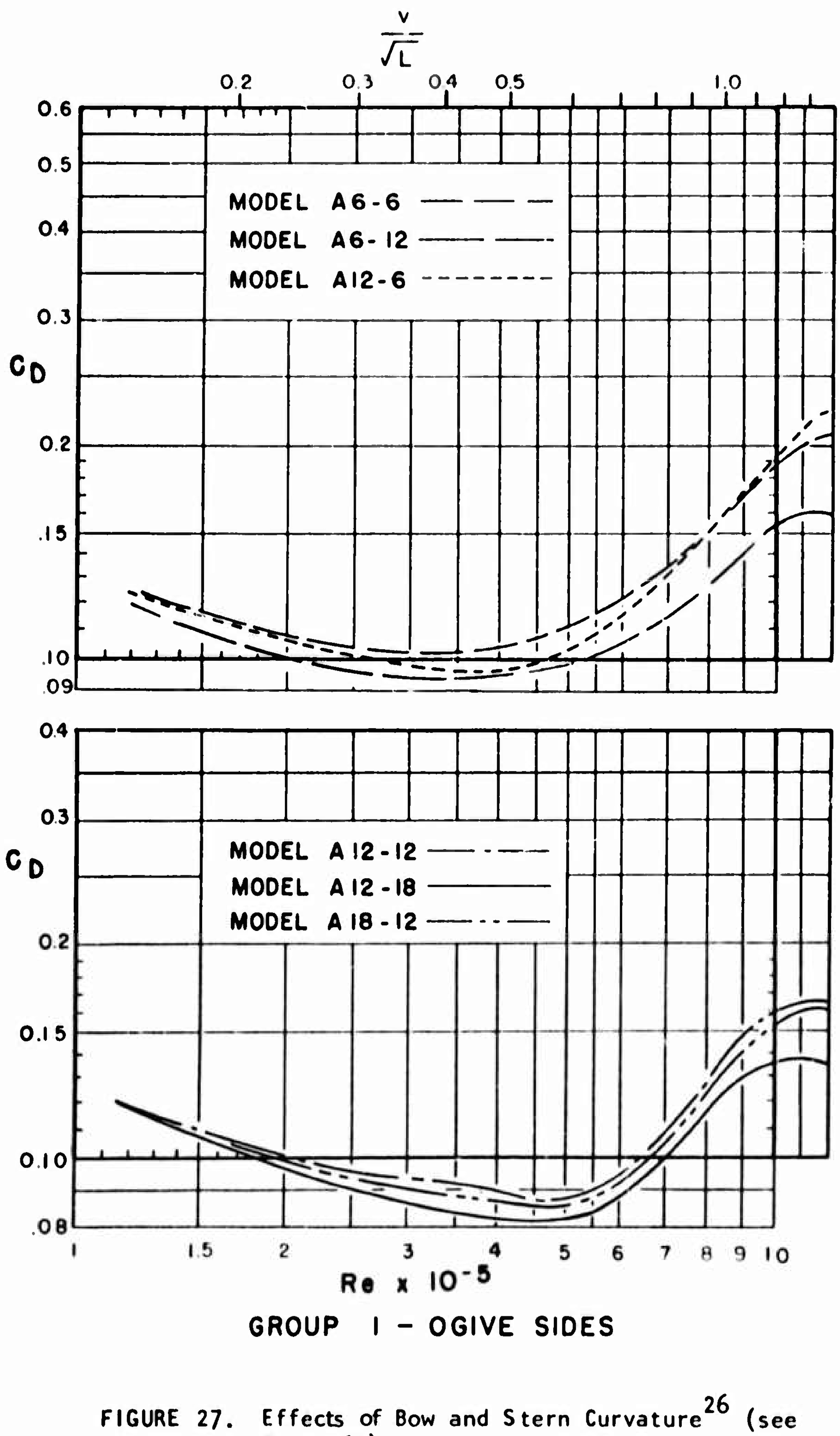
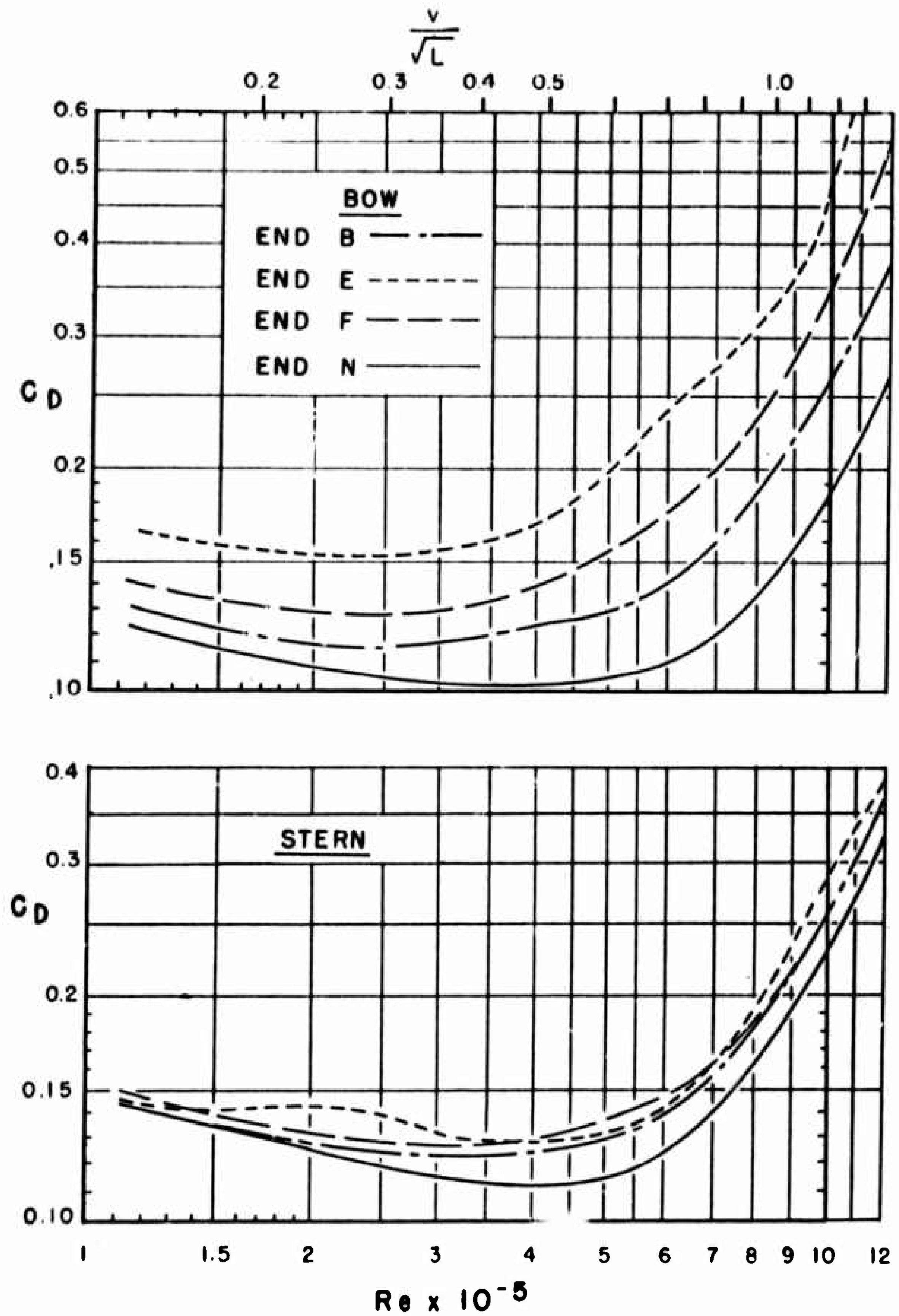
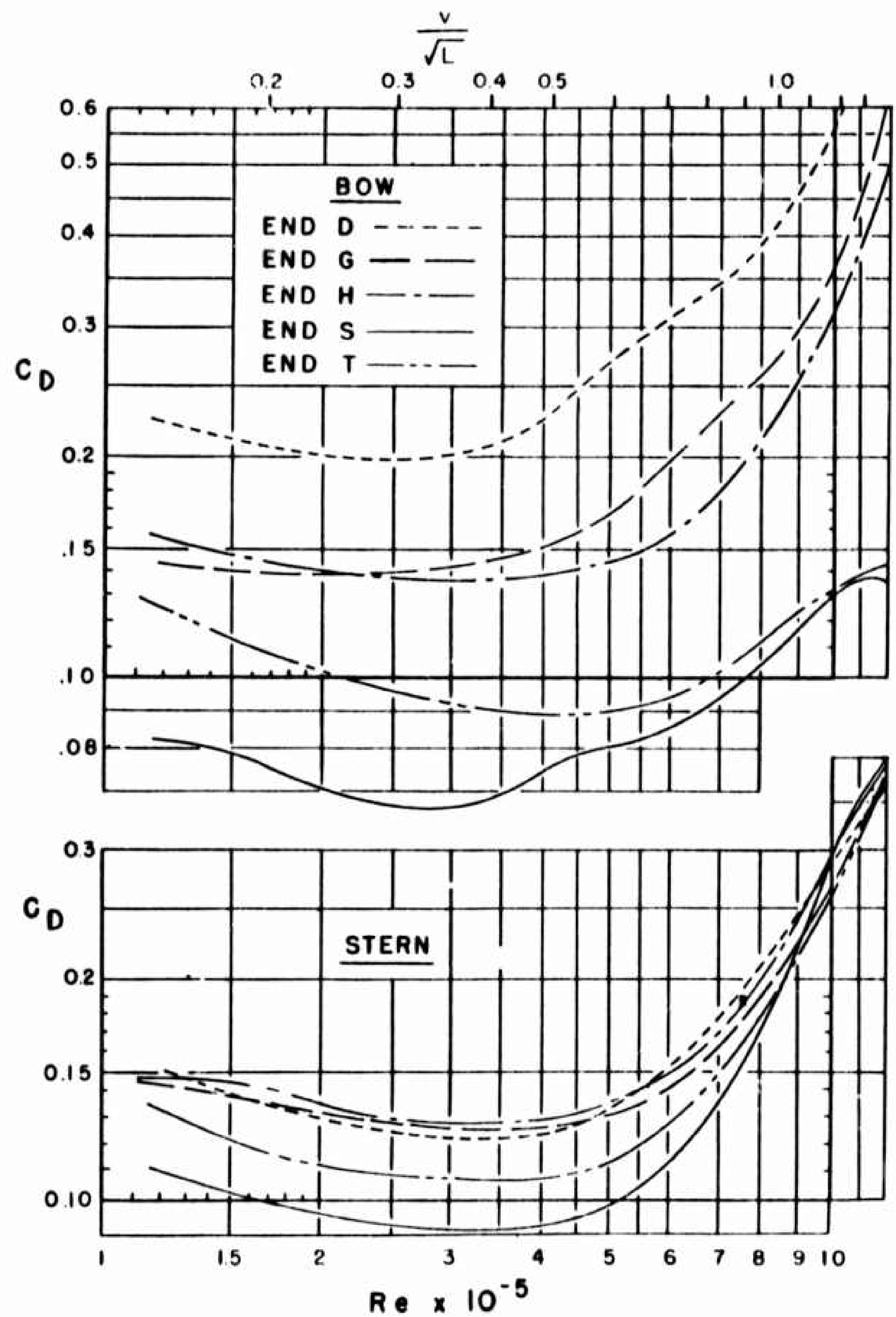


FIGURE 27. Effects of Bow and Stern Curvature[26] (see Fig. 26D)



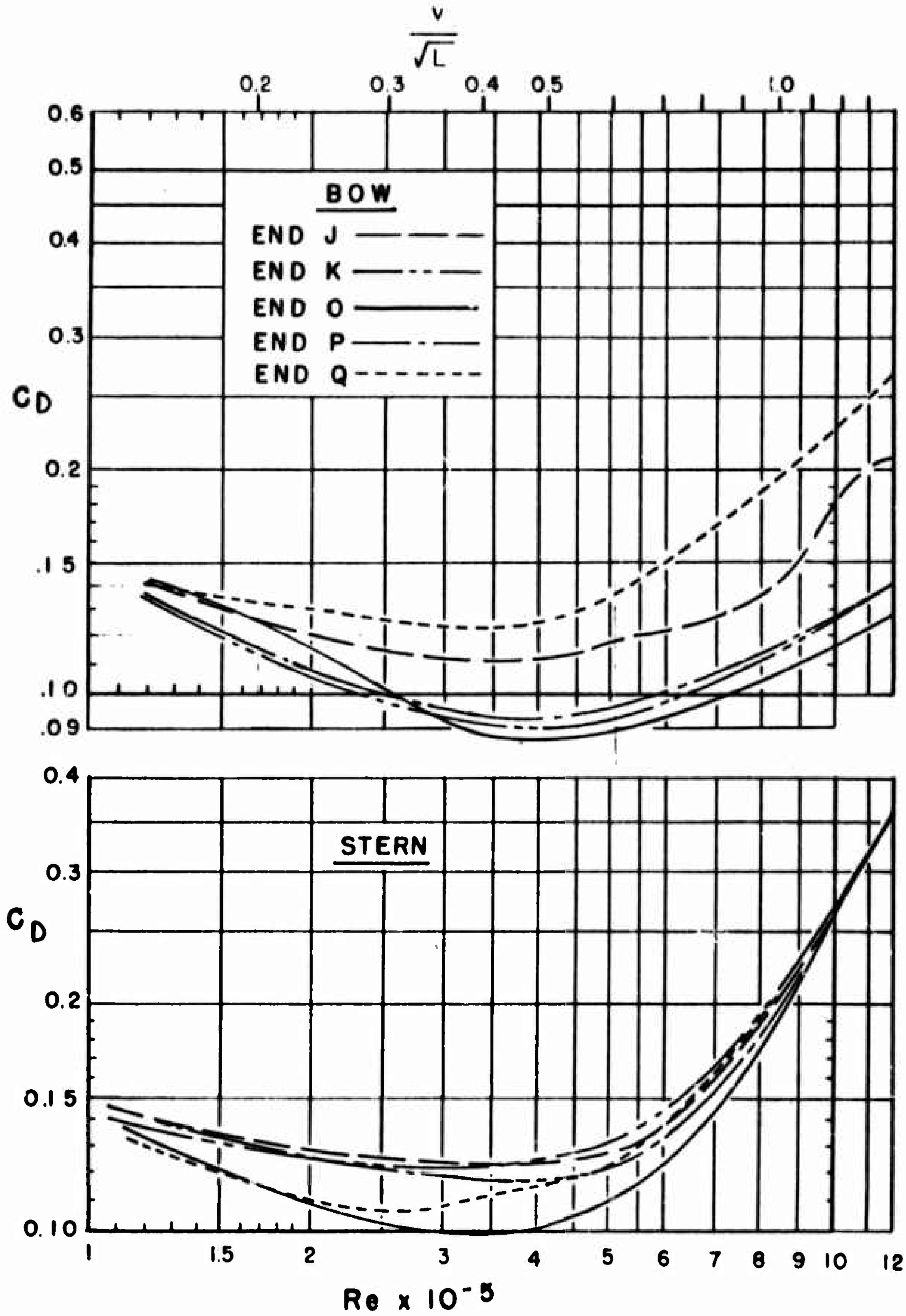
GROUP 2 - RADIUS SIDES

FIGURE 28. Effects of Bow and Stern Side Radius[26]
(see Figs. 26A and 26C)



GROUP 3 - MITERED SIDES

FIGURE 29. Effects of Bow and Stern Side Treatment[26]
(see Figs. 26A, B, C, and D)



GROUP 4 - BEVELED BOTTOM

FIGURE 30. Effects of Bow and Stern Beveled Bottom[26] (see Figs. 26B and 26C)

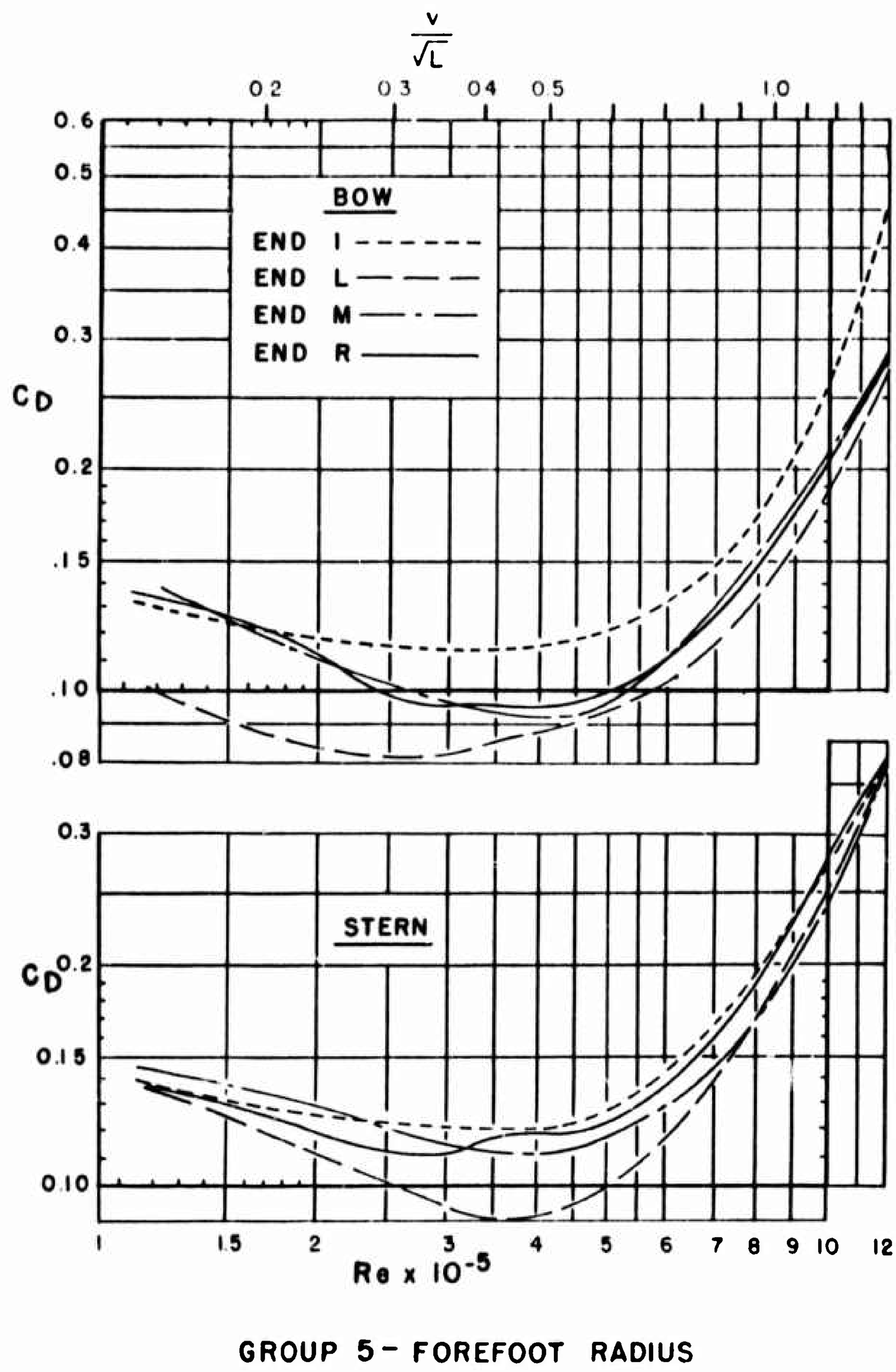


FIGURE 31. Effects of Bow and Stern Forefoot Treatment[26] (see Figs. 26B and 26C)

Chapter 3
PROPULSION

This section is concerned with the approach and language of the naval architect in dealing with propulsion devices. Discussion will be restricted to mechanical devices as opposed to sails, and will cover only those devices acting on the water.

Self-propulsion of any kind is necessarily a reaction phenomenon, with the propelling force derived from a pulling or a pushing on some external matter. For the self-propulsion of bodies in water, the needed propulsion force is obtained by changing the momentum of a mass of fluid so that it acts in a direction opposite to the desired direction of vehicle motion. In practice, the accelerating device is usually a propeller, pump, or paddle.

SCREW PROPELLER

For more than a century, the screw propeller has reigned supreme as the primary method of ship propulsion, because of its high propulsion efficiency, its simplicity, and its relatively small size. The blade-element theory of the screw propeller treats the blade sections as foils, computes the lift and drag forces, and resolves these two forces into the thrust and torque. Performance depends on the angle of attack of the blade section relative to the incoming fluid, and the angle of attack in turn depends on rotational speed, pitch, and velocity of advance. Summation of the thrust and torque over all the differential blade elements yields the full propeller performance.

The number of blades for ship propellers varies from two to six, but as a rule is three or four; choice is dictated by many considerations. In general, the optimum diameter of the screw increases with decreasing number of blades. The maximum propulsive efficiency η_p also increases with decreasing number of blades. Propeller vibrations, however, are reduced somewhat as the number of blades increases.

The selection of a screw for a given application involves the consideration of many trade-offs in weight, efficiency, cost, size, etc., and there are operating limitations associated with cavitation and ventilation. In ship design, the hub of the propeller is, ideally, located at least one diameter below the free surface of the water, to avoid ventilation (the sucking in of air from the surface), or to prevent the blade tip from breaking the surface when the wake trough forms behind the ship. Cavitation is the formation of water-vapor bubbles or cavities at locations on the leading face of the propeller where the dynamic pressure falls below the fluid vapor pressure.

Definition of Terms

The approach presented here is considered standard for the naval architect and is taken from the definitive work of Van Manen.[29] It is basically a definition of the manner in which the hull, the propulsor, and the interference effects are accounted for in the determination of self-propelled ship performance. The discussion is centered on the screw propeller, but the approach is applicable to all propulsion devices. The reader, however, must unfortunately bear with a number of preliminary definitions.

The <u>indicated horsepower</u> (ihp) is that power developed by the engine.

The <u>shaft horsepower</u> (shp) is taken as the power delivered to the shaft by the main engines, aft of the gearing and thrust block.

The <u>delivered horsepower</u> (dhp) is that power actually delivered to the propeller, to propel the ship:

$$\text{dhp} = \frac{2\pi Q n}{550} \tag{36}$$

where Q = torque at the propeller, ft-lb

n = shaft revolutions per second

Because of friction losses in bearings, thrust blocks, stuffing boxes, and transmission gear, the propelling power supplied to the propeller is

naturally somewhat less than the power supplied by the propelling machinery.

The power required to overcome the resistance of the hull is called the effective horsepower (ehp).

$$\text{ehp} = \frac{R_T \cdot v_s}{550} \tag{37}$$

where R_T = total hull resistance, lb, at forward ship speed (v_s), ft/sec

The efficiency of an operation is generally defined as the ratio of useful work or power obtained to the work or power used to carry out the operation. Such propulsive efficiency in a ship would be represented by the effective horsepower divided by the indicated horsepower. However, mechanical efficiencies, gear losses, and transmission losses vary according to type of machinery and layout, and so an over-all efficiency is actually rather difficult to define. Hence the following, more meaningful, measures of propulsive efficiencies are used:

The quasi propulsive coefficient, η_D, is represented by

$$\eta_D = \frac{\text{ehp}}{\text{dhp}} = \frac{R_T v_s}{2\pi Q n} \tag{38}$$

The transmission efficiency, η_s, is represented by

$$\eta_s = \frac{\text{dhp}}{\text{shp}} \tag{39}$$

(This is usually in the region of 90 percent for vehicles.)

The propulsive efficiency, η_p, as used in this text, is represented by

$$\eta_p = \frac{\text{ehp}}{\text{shp}} = \eta_D \cdot \eta_s \tag{40}$$

Simple momentum considerations yield the following expression for thrust:

$$T = \rho \, q \, \Delta v \tag{41}$$

where T = magnitude of thrust vector

ρ = fluid mass density

q = volume flow of fluid

Δv = change in velocity of fluid parallel to the vector T

The <u>ideal efficiency</u>, η_I, of any such propulsive device may also be calculated, from

$$\eta_I = \frac{v_A}{v_A + \frac{\Delta v}{2}} \tag{42}$$

where η_I = ideal efficiency of the propulsive device

v_A = initial velocity of the fluid under consideration with respect to propeller

Δv = change in velocity imparted to the fluid

From Eq. (42) when $v_A = 0$ (i.e., when the vehicle is not moving), efficiency is obviously zero. Not so obvious, however, is the fact that the maximum obtainable efficiency is greatest when Δv is small; that is, η_I decreases with increasing Δv. Thus Eqs. (41) and (42) clearly show that for most efficient thrust production it is better to accelerate a large quantity of water a little than to accelerate a small quantity of water a great deal. This principle, translated into hardware terms, explains the higher efficiencies obtainable with propulsion systems of larger capacity.*

*And, conversely, the low efficiencies generated by high-velocity jet pumps.

The propeller in open water with a uniform inflow velocity at a speed of advance v_A has an open water efficiency, η_o.

$$\eta_o = \frac{\text{work out}}{\text{work in}} = \frac{T \cdot v_A}{2\pi Q n} \tag{43}$$

In reality, the presence of the hull distorts the flow to the propulsor and the propulsor distorts the flow around the hull. Thus v_A, the velocity of the fluid approaching the propulsor, does not necessarily equal v_s, the velocity of the vehicle. The performance of practical importance, however, is that of the entire self-propelled vehicle. This performance depends on the hydrodynamic drag of the hull, the open-water performance of the propulsor, the effect of the propulsor on hull drag, and the effect of the flow around the hull on the propulsor. As a result of the mutual interference between hull and propeller, the propulsive coefficient, η_D, is generally not equal to the open-water propulsive efficiency η_o of the propeller. Likewise, the open-water torque, Q_o, required to turn the propeller, is different from that (Q) required at the propeller shaft of the vehicle.

It is important to remember that the speed v_A of Eq. (43) is the relative velocity of the water particles behind the ship, in way of the propeller. Owing to the usual widening of the streamlines near the stern, to the friction along the hull, and to the wave system of the vehicle, this intake velocity (v_A) of the water entering the screw is generally less than the forward speed of the ship, v_s.

The absolute velocity of advance, which is the difference between the ship speed (v_s) and this intake velocity (v_A), is called the <u>wake speed</u>, v_W.

$$v_W = v_s - v_A \tag{44}$$

It is common practice to express wake speed (v_W) in relation to ship speed as the <u>wake fraction</u>, w, defined by Taylor as

$$w = \frac{v_w}{v_s} = \frac{v_s - v_A}{v_s} = 1 - \frac{v_A}{v_s} \tag{45}$$

$$v_A = v_s\,(1-w) \tag{46}$$

The thrust , T, of the propeller, however, generally exceeds ship towing resistance R_T . The difference is in part due to the increase in frictional resistance caused by the screw's acceleration of the water near the stern. Also, the stern wave system of the ship may, in special cases, be influenced by the propeller and, in turn, cause a change in wave-making resistance. This difference is called <u>thrust deduction</u> and is generally expressed as a fraction, t , of the thrust.

$$t = \frac{T-R_T}{T} = 1 - \frac{R_T}{T} \tag{47}$$

where t = the <u>thrust deduction factor</u>.

The propulsive coefficient η_D can now be expressed as the product of three factors.

$$\eta_D = \frac{R_T v_s}{2\pi Q n} = \left[\frac{T \cdot v_A}{2\pi Q n}\right]\left[\frac{R_T v_s}{T \cdot v_A}\right]\left[\frac{Q_o}{Q}\right] \tag{48}$$

The first factor represents, from Eq. (43), the propulsive efficiency η_o , of the propeller in open water.

The second factor, which is the ratio of the effective horsepower (Eq. [37]) to the propulsive power ($T \cdot v_A$) of the screw, is called the <u>hull efficiency</u>, η_H . From Eqs. (46) and (47), this ratio may be rewritten as a function of the wake fraction and the thrust deduction factor, and is a measure of the mutual interference of propeller and hull.

$$\eta_H = \frac{R_T v_s}{T \cdot v_A} = \frac{1-t}{1-w} \tag{49}$$

The third factor represents the ratio between the open-water torque, Q_o , and the actual delivered torque, Q . This difference is caused by the turbulence and inequality of the flow field behind the hull and near the rudder. It is called the <u>relative rotative efficiency</u>, η_R .

$$\eta_R = \frac{Q_o}{Q} \tag{50}$$

The interference effects between hull and propeller (wake fraction w , thrust deduction t , and relative rotative efficiency η_R) depend on the hull form and the position of the propeller with respect to the hull. The experienced naval architect can make an estimate of these terms, based on many earlier ships of similar design. The automotive-vehicle designer, however, needs general guidelines. The interference terms are sensitive to the positioning of the propeller relative to the hull. If the propeller is situated well below the hull, losses due to turbulence, ventilation, and cavitation will be small, and the wake speed v_w will also be small, with the result that hull and relative rotative efficiencies, η_H and η_R , will be high. On the other hand, if the prop is placed immediately behind a flat stern, large amounts of energy will be expended just to bring water to the propeller; therefore the wake speed v_w will be high and the advance coefficient J (Eq. [52]) low. This will result in a low propeller efficiency (η_o) and a low hull efficiency (η_H). In terms of hull efficiency, high values of wake fraction (w) are desirable. However, high wake fractions yield low advance coefficients (J) which usually result in low propeller efficiency (η_o).

The wake fraction may vary from 0.1 on well-designed ships to 0.5 on poorly designed ones. In general, most military amphibious vehicles are considered poor ships, and a reasonable approximation for w would probably be somewhere between 0.3 and 0.4. A conventional ship hull will have a thrust deduction factor (t) between 0.1 and 0.4. Barges, tugs, and boxy hulls will have thrust deduction factors near 0.5. Well-designed hulls will have relative rotative efficiencies (η_R) close to 1.0. Barges, tugs, and other box-like hulls, however, may have η_R as low as 0.9. An estimate of 0.9 for military amphibious vehicles, therefore, will in general be not too much in error.

When Eqs. (43), (49), and (50), are combined, the propulsive coefficient (Eq. [48]) becomes

$$\eta_D = \eta_o \cdot \eta_H \cdot \eta_R \tag{51}$$

which is the more convenient form commonly used in naval architecture.

Propeller performance is usually presented on charts (Fig. 32), in non-dimensional form as a function of the advance coefficient, J, which is defined as

$$J = \frac{v_A}{nD} \tag{52}$$

where D = propeller diameter

If a propeller moved with the same action as a screw in a nut, it would in one revolution move forward a distance equal in magnitude to its pitch. Because of the acceleration of the water and the drag of the vehicle, however, the real propeller actually moves forward something less than its pitch, in one revolution. This difference in forward motion is referred to as <u>slip</u> and is used in ratio form S_R.

$$S_R = 1 - \frac{Pn - v_A}{Pn} = 1 - \frac{v_A}{Pn} \qquad (53),(54)$$

where P = pitch

The numerator of the right-hand side of Eq. (53) is called the <u>slip velocity</u> and is the difference between the <u>pitch velocity</u>, Pn, and the speed of advance of the screw, v_A (see Fig. 33). Since, in Eqs. (53) and (54), v_A is the speed of advance of the screw with respect to the surrounding water, this slip, S_R, is called the <u>real slip</u>. The slip based on ship speed, v_s, is called the <u>apparent slip</u>, S_A.

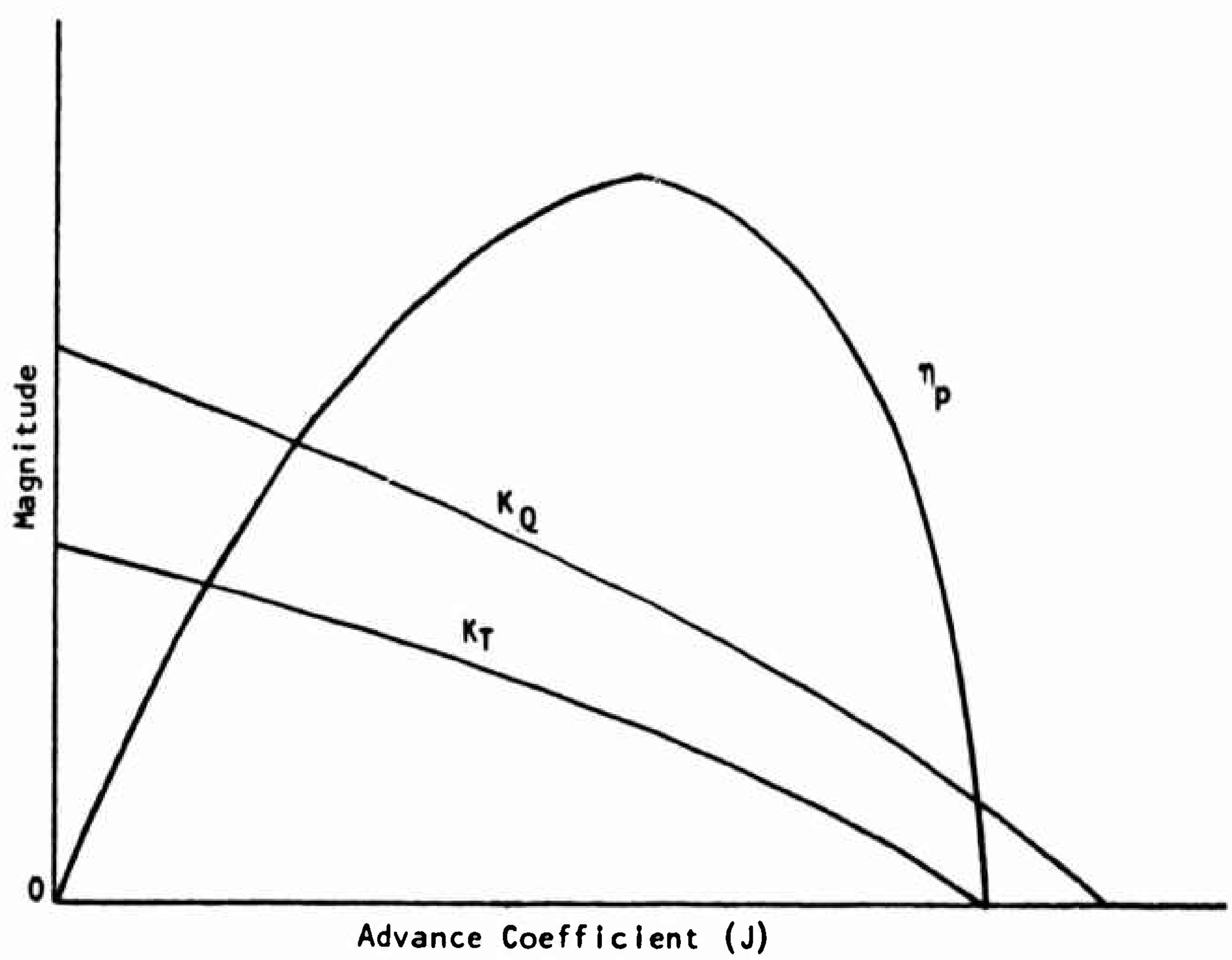


FIGURE 32. Typical Performance Characteristics of a Screw Propeller of Known Shape and Pitch/Diameter Ratio.

$$S_A = \frac{Pn - v_s}{Pn} = 1 - \frac{v_s}{Pn} \qquad (55)$$

The ratio between the two slip values and the wake fraction (see Fig. 34) is represented by

$$\frac{1 - S_R}{1 - S_A} = \frac{v_A}{v_s} = 1 - w \qquad (56)$$

The relationship between the apparent slip, S_A , and the advance coefficient, J , is

$$J = (1 - S_A)\,\frac{P}{D} \qquad (57)$$

Propeller thrust and torque are usually presented on charts (see Figs. 32 and 35*) in terms of non-dimensional <u>thrust coefficient</u>, K_T , and <u>torque coefficient</u>, K_Q .

$$K_T = \frac{T}{\rho\, D^4 n^2} \qquad (58)$$

$$K_Q = \frac{Q}{\rho\, D^5 n^2} \qquad (59)$$

The left side of Fig. 32 and of Fig. 35 denotes the area of high screw-loading (towing). The condition when the advance coefficient, J , is zero, or 100-percent slip, represents the so-called "bollard pull" condition and produces maximum thrust.

The propeller efficiency may be calculated in terms of K_T , K_Q , and J , from

$$\eta_o = \frac{K_T}{K_Q} \cdot \frac{J}{2\pi} \qquad (60)$$

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*Figure 35 is reproduced from a naval architect handbook and contains symbols somewhat different from those used in this report.

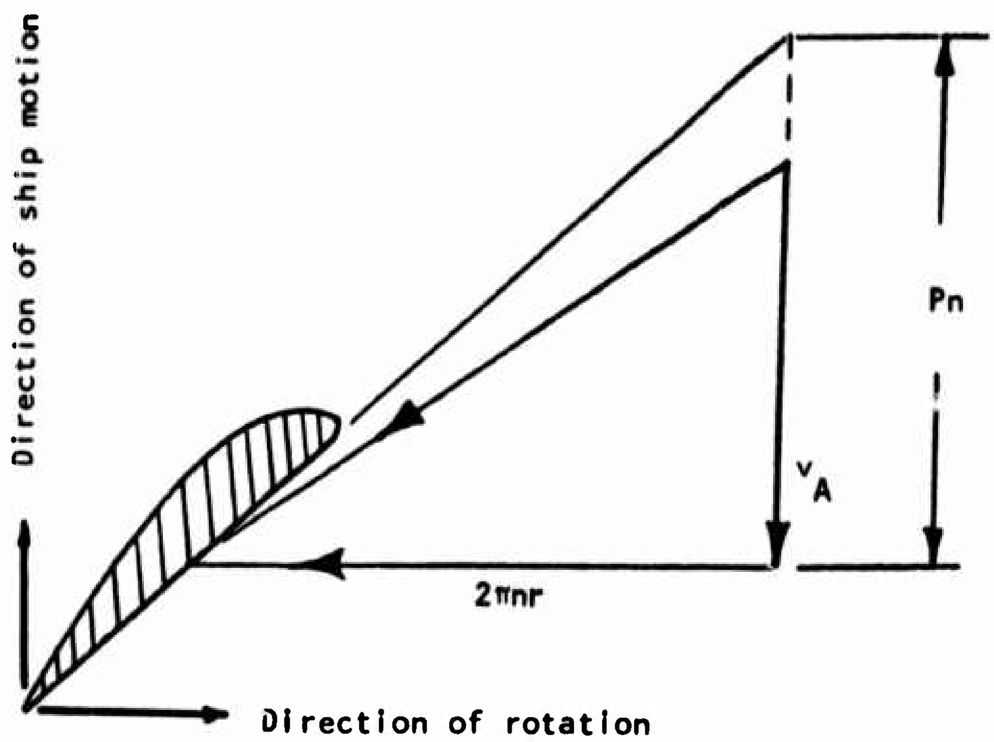

FIGURE 33. Relation Between Pitch Velocity (Pn), Speed of Advance (v_A), Slip Velocity ($Pn-v_A$), and Blade Tangential Velocity at Radius r.

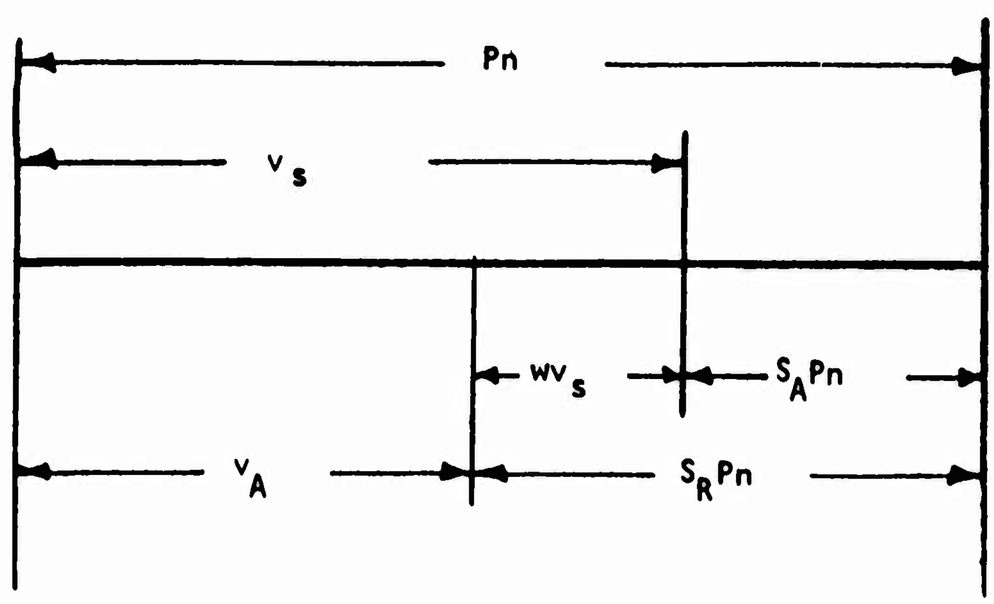

FIGURE 34. Relation Between True Slip (S_R), Apparent Slip (S_A), and Wake Speed (v_w) (equal to the product of the wake fraction, w, and the ship speed, v_s).

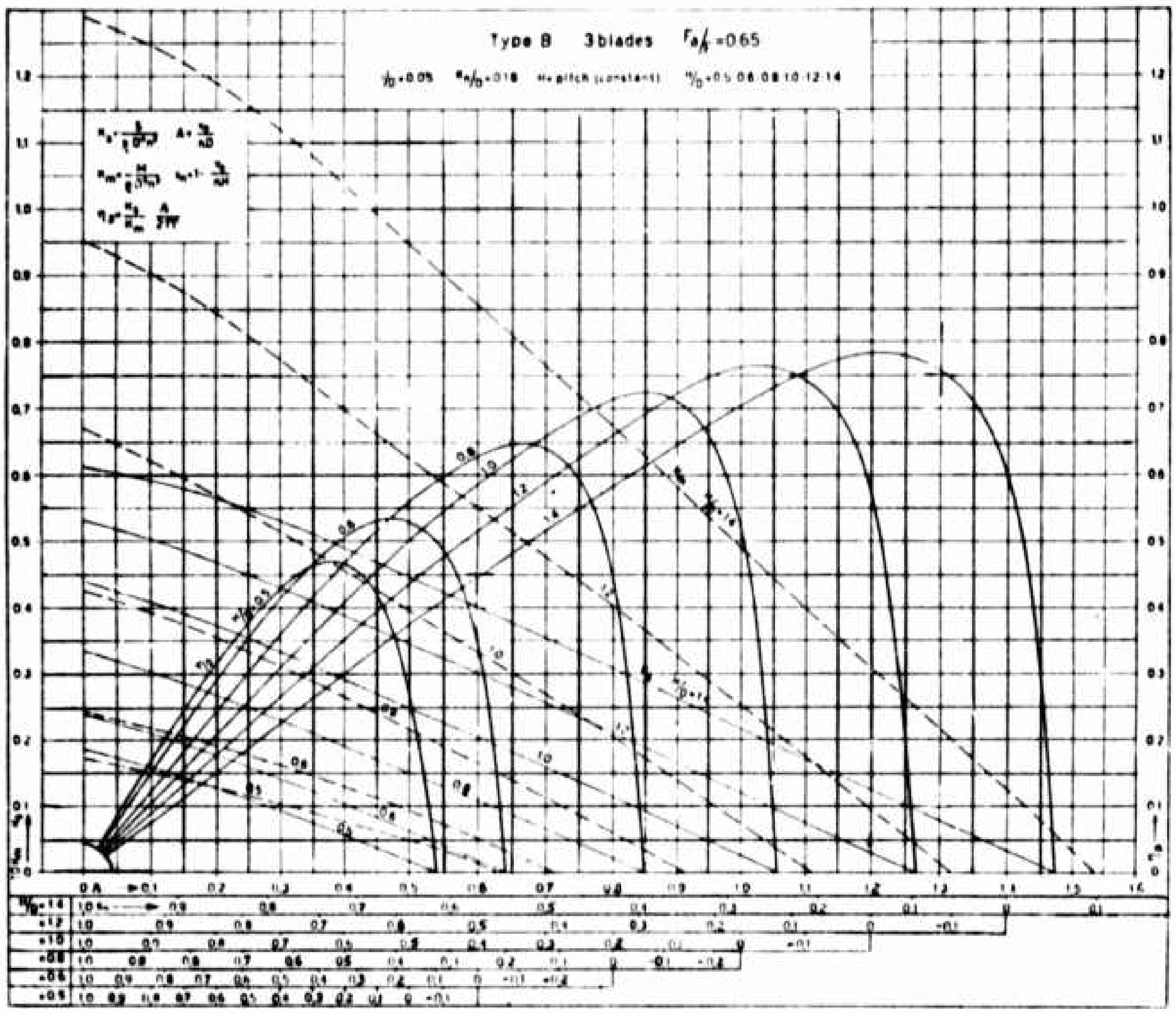

FIGURE 35. Results of Open-Water Tests on Three-Bladed Propellers of NSMB B-365 Type From Ref. 33.

Figure 32 is typical of the form in which the performance of a propeller is presented. Since all of the parameters are dimensionless, the laws of geometric similitude apply, and the performance of all similar propellers based on this geometry can be readily calculated.

Cavitation reduces the quality of propeller performance and is deleterious to the blades in that erosion quickly occurs. Cavitation can be avoided if operating conditions are limited. Figure 36 shows the cavitation criteria for all screw propellers, and makes it possible to determine safe operational limits. The diagram uses the cavitation number, σ , defined as

$$\sigma = \frac{P_o - P_v}{\frac{\rho}{2} (v_A)^2} \tag{61}$$

where pressure burden = total pressure - vapor pressure

= atmospheric pressure + water pressure - vapor pressure

= $P_o - P_v$

$\approx$ 2064 + $\rho g d$

(d = depth of submergence, ft, of propeller hub)

Propeller Selection

A screw series is comprised of models whose characteristics, such as pitch/diameter ratio, number of blades, blade outline, shape of blade sections, and blade thicknesses, are varied systematically. The best-known screw series are those designed by Froude, Schaffran, Taylor, Schoenherr, Gawn, and Troost. It is much easier to select a propeller from an existing screw series than to initiate a new design. Consequently, propeller selection rather than propeller design will be discussed here.

The vehicle designer usually starts with a hull for which he requires propeller and powerplant to obtain a specified speed. His first step, therefore, is to conduct model tests, to determine the plot of hull resistance (R_T) versus speed (v_s). Figure 37 is an example of such a plot (see also Figs. 15, 17, 18, and 19). The following outline can then be used to select a suitable propeller from series data:

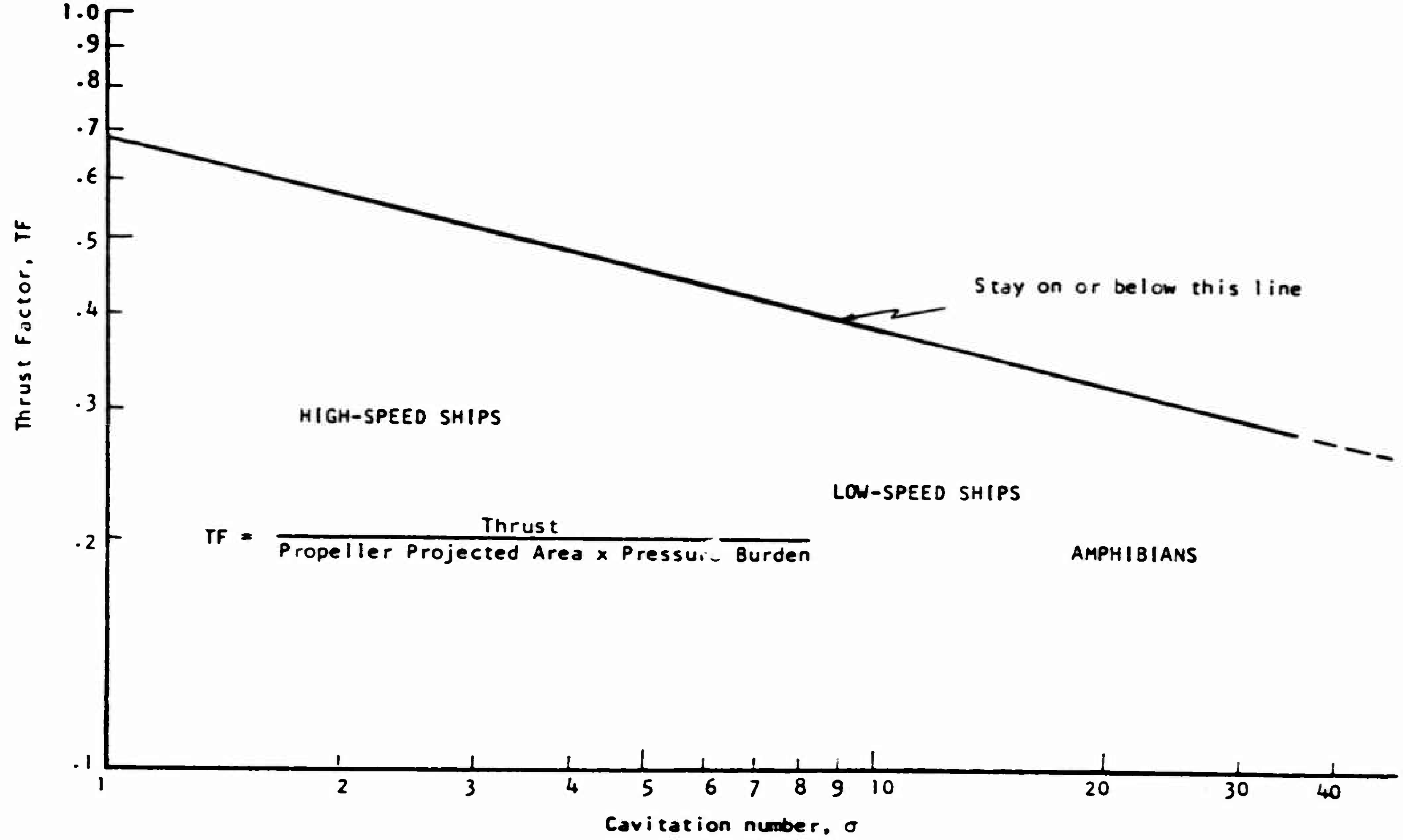

FIGURE 36. Wageningen Tank Curve of Cavitation Limits

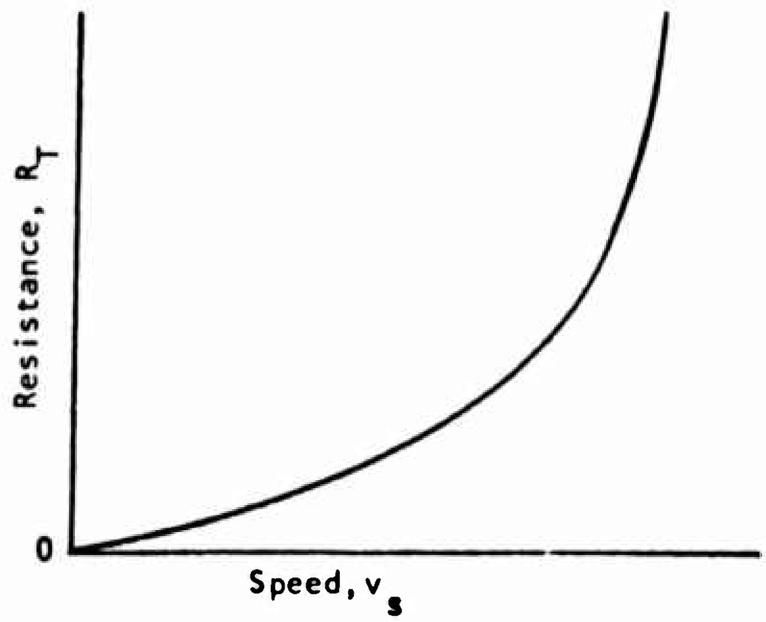

FIGURE 37. Typical Resistance R_T Versus Hull Speed v_s

(1) Select a propeller diameter, D, which will fit within the confines of space imposed by the geometry of the hull and the required draft.

(2) Choose a propeller speed, n, consistent with the optimum operating characteristics of the engine and transmission system.

(3) Using an estimated thrust deduction factor, t (near 0.5 for amphibians), and the estimated hull drag, R_T, at the desired speed, v_s, calculate the required thrust, T, from Eq. (47).

$$T = \frac{R_T}{1 - t} \tag{62}$$

(4) Calculate the required thrust coefficient, K_T (Eq. [58]).

$$K_T = \frac{T}{\rho \, D^4 n^2}$$

(5) Using an estimated wake fraction, w, (between 0.3 and 0.4 for most military amphibians), calculate the intake velocity, v_A, from Eq. (56).

$$v_A = v_s (1 - w) \tag{63}$$

(6) Determine the advance coefficient, J (Eq. [52]).

$$J = \frac{v_A}{nD}$$

(7) On the appropriate propeller-series chart (like Fig. 35), enter the calculated values of K_T and J, and select the pitch/diameter ratio, P/D, giving maximum propeller efficiency, η_o.

(8) Note the corresponding value of K_Q and determine the open-water torque, Q_o, from Eq. (59).

$$Q_o = K_Q \cdot \rho\ D^5 n^2 \tag{64}$$

(9) Using an estimated value of relative rotative efficiency, η_R (a reasonable approximation of 0.9 can be used for military amphibious vehicles), calculate the required propeller torque Q , from Eq. (50).

$$Q = \eta_R\ Q_o \tag{65}$$

(10) Then calculate the required developed horsepower (Eq. [36]).

$$dhp = \frac{2\pi Qn}{550}$$

(11) Use Fig. 36 to check for cavitation. If the solution lies below the line, the selection should be satisfactory; if it does not, a different propeller must be selected. Try another diameter, another series, or another operating speed.

Several iterations of the above calculations may be necessary to match vehicle drag, propeller thrust, and engine torque-horsepower in an optimum relationship.

KORT NOZZLE

An open propeller enclosed in a shroud becomes a Kort nozzle propulsor. The Kort nozzle (Fig. 38) is essentially a thrust-improvement device. It is designed to --

(1) Increase thrust capability.

(2) Obtain better cavitation performance.

(3) Reduce propeller size (as compared with an open propeller).

These advantages are not all simultaneously realized, since shroud shape is selected to attain one of these features.

The nozzle achieves its affect by helping the water to flow smoothly into the propeller blades. By reducing the radial flow off the tips of

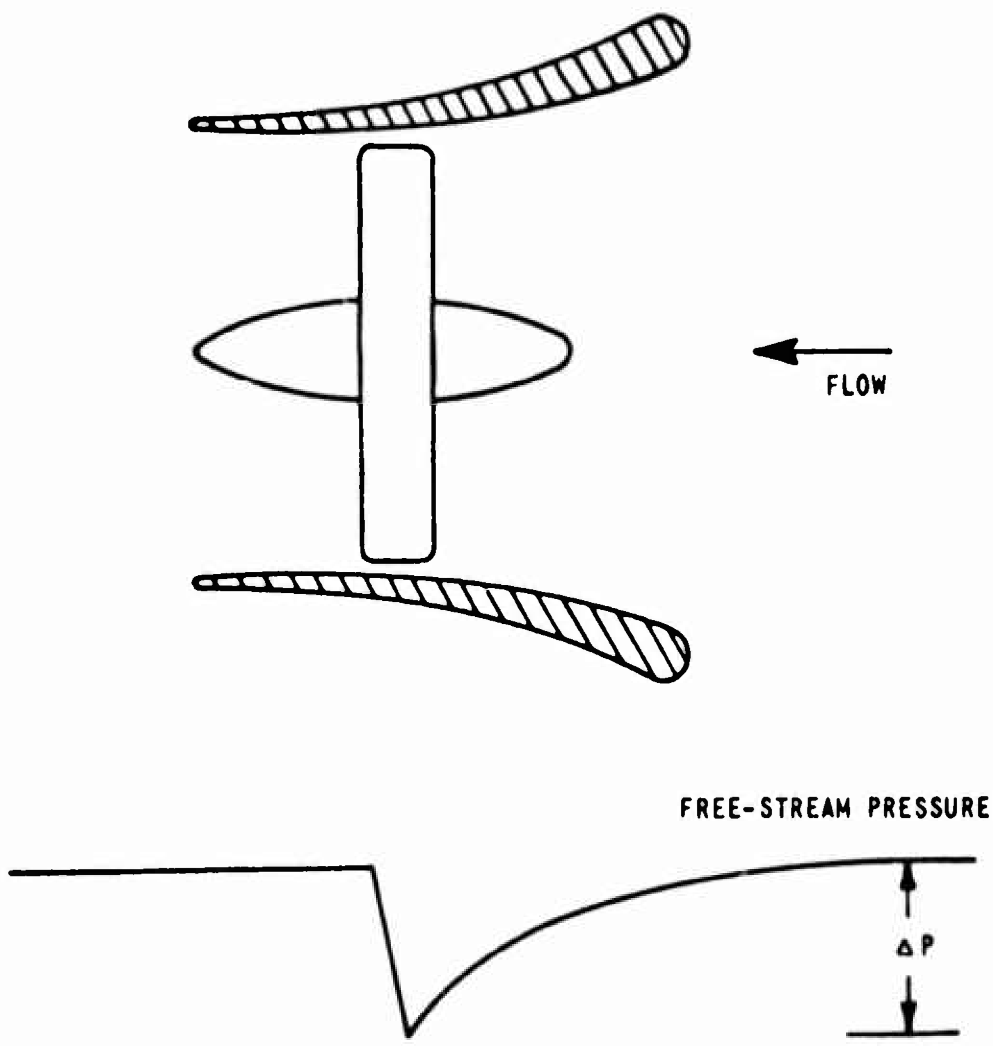

FIGURE 38. Schematic of Kort Nozzle

the propeller, it also makes the blade act as a more efficient high-aspect-ratio foil. Thus, to be effective, clearance between the blade tips and the inside of the nozzle must be small.

The Kort nozzle is especially attractive for amphibious vehicles, since the shroud also provides a measure of protection for the propeller blades. As a rule, the slight penalty in increased hydrodynamic resistance easily justifies the employment of this device. It has recently been reported that a Kort-nozzle shrouded propeller, mounted to the drive sprocket, significantly improved the water speed of a tracked vehicle.

WATER JETS

Water jets (hydrojets or pump jets) are devices which take in water, raise its pressure, and eject it rearward at a higher velocity than it had when it entered. Figure 39 shows a typical arrangement. Particular note should be made of the comparatively large vehicle volume which is filled with water. The loss of buoyancy is frequently forgotten in the design of such systems, with obviously detrimental effects.

Since water jets usually operate with modest flows and large velocity changes, their propulsion efficiencies are inherently lower than those of corresponding screw propellers or paddle wheels (see Eqs. [36] and [37]). Fluid flow losses due to high flow velocities in the internal ducting of the water-jet device further reduce efficiency. Of particular interest are the losses at the inlet and the effect on such losses as ship speed increases. These losses will increase for an inlet such as is shown in Fig. 39. But if the inlet were oriented so that the incoming fluid impinged on the opening, the inlet loss would decrease, and performance would improve, since some of the kinetic energy of the fluid entering the duct would be recovered.

The advantages of the water jet, as compared with the screw propeller, are listed below.

(1) There need be no projections outside the hull.

(2) Steering and reverse thrust can be obtained by swiveling the exit nozzle, thereby eliminating rudders and reverse gears.

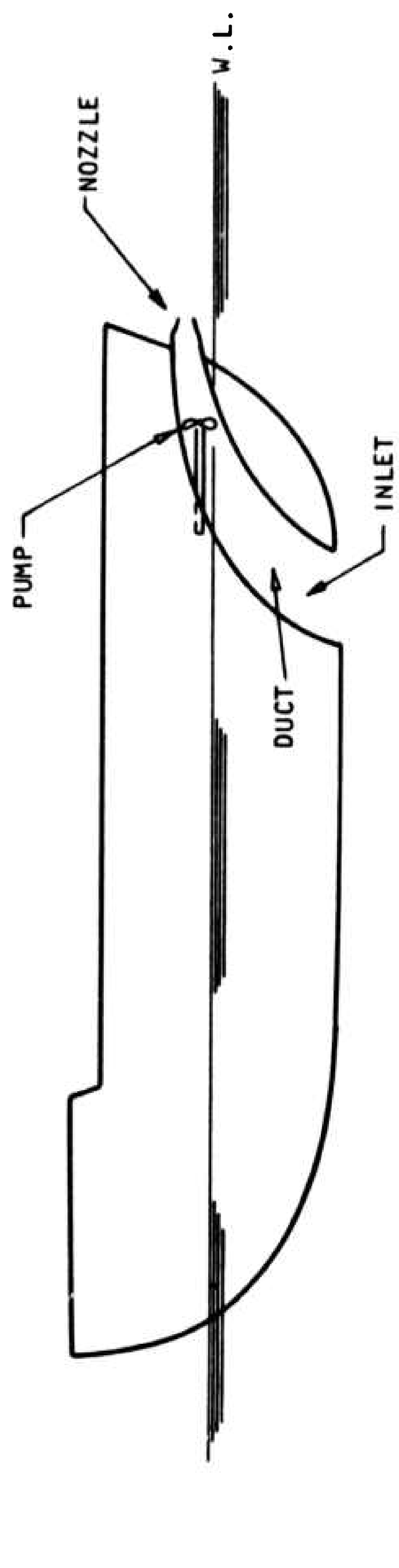

FIGURE 39. Schematic Arrangement of Water-Jet-Propelled Craft

(3) There is less chance of cavitation in a properly designed water jet.

(4) Water may be taken in from regions of high static pressure, thereby reducing effective wave drag.

The disadvantages of the water jet are:

(1) Special care must be taken to prevent ingestion of debris by the pump.

(2) The buoyancy of the hull is reduced by the volume of the ducting.

(3) Lower propulsion efficiencies are obtained than with a screw propeller.

A sample of manufacturer's performance data for typical jet pumps is presented in Figs. 40, 41, and 47.

An extensive analysis of pump-jets and their performance has been performed by Henderson et al.[30] This study, however, is somewhat controversial and has frequently been attacked; but no other comprehensive analysis is presently in print. For rough calculations of amphibious-vehicle performance, the designer should therefore employ data supplied by the manufacturer, and apply it to predicted hull-drag (as explained earlier in this paper). An important note: Manufacturers like to show their devices to best advantage; therefore they frequently supply data for thrust-versus-horsepower curves which are measured at zero forward speed (Fig. 47), when the values are largest. This thrust will fall off considerably with increasing speed (Fig. 40).

WHEEL PROPULSION

Amphibious wheeled vehicles have, for some time, been able to propel themselves simply by spinning their wheels. When the top of a wheel is above water, the tire acts somewhat as a paddle wheel. It has been observed, however, that propulsion is also obtained when the wheel is totally submerged. The exact mechanism of this propulsion is not yet known, but studies are presently being conducted to investigate the phenomenon.

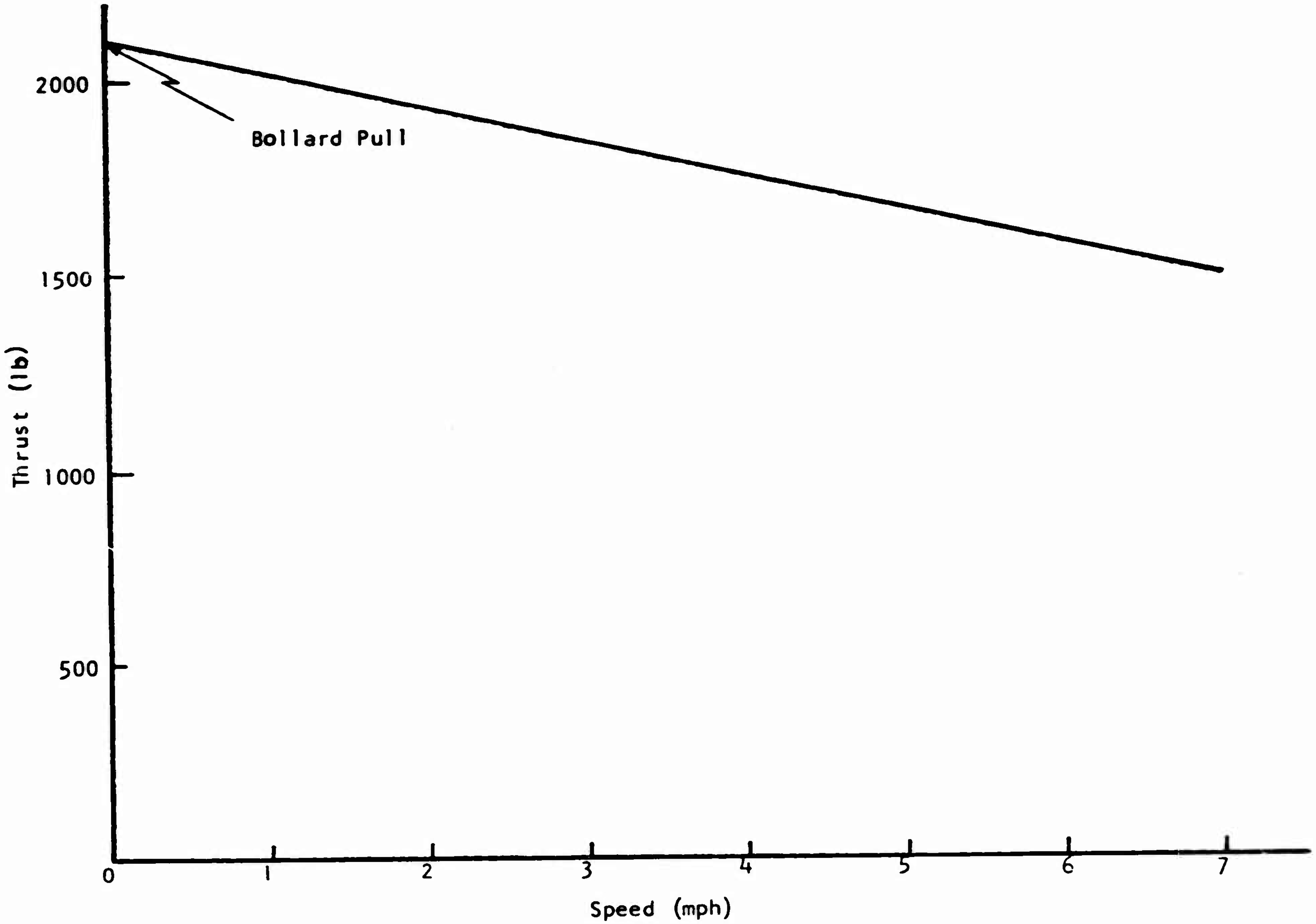

FIGURE 40. Fall-Off of Thrust with Speed of a Typical Jet-Pump (at Constant Horsepower)

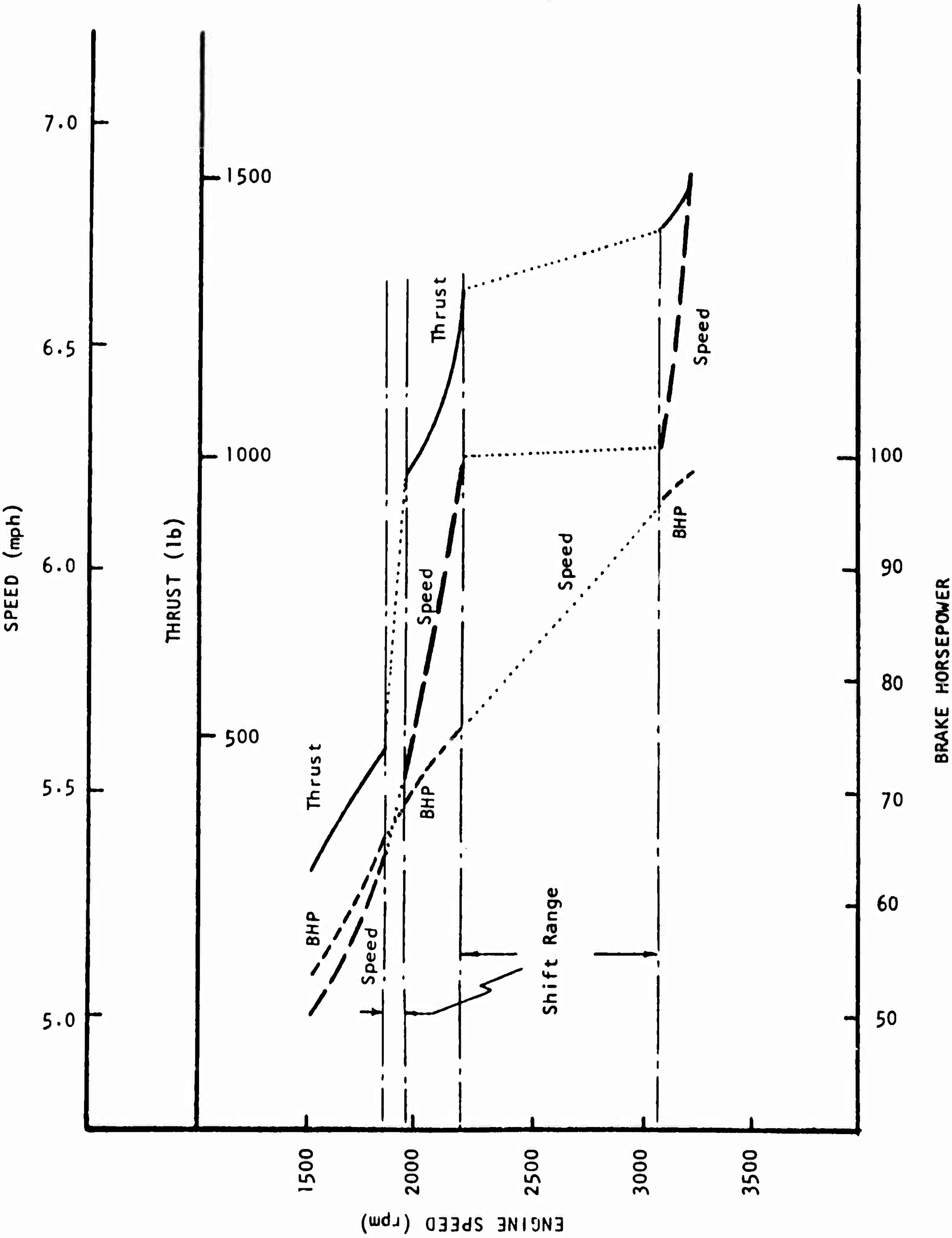

FIGURE 41. Performance Curve of a Jet-Pump Attached to the XM-147 Superduck (Courtesy of Hanley Hydrojet)

Spinning wheels propel vehicles just as tracks do, but are even less efficient. Like tracks, they can also be employed as a means of water propuls' for a vehicle which is to be used predominantly in land operation. Their advantage lies not in their efficiency, but in the fact that they are already required for land operations.

Performance is affected by such factors as tire tread, diameter, and width; and by the juxtaposition and geometry of fenders, deflectors, and baffle plates. Rymiszewski[31] conducted tests on a 1/8-scale self-powered model of an 8 x 8 5-ton cargo truck. These tests indicated that wheel-shroud design could increase water speed up to 70 percent above the best unshrouded condition, and that improper wheel-shrouding and/or worn tire-treads might decrease the propelling forces of the wheels. Similar tests conducted at the Davidson Laboratory[32,33] indicate propulsive coefficients in the order of 0.5 to 2.0 percent. Speeds of up to 2 mph have been obtained in this manner.

A recently proposed, though as yet untried, method of wheel propulsion that appears to have promise is the use of the disc portion of the wheel as an axial or centrifugal flow pump, to create inboard flow about the wheel hub.[33] Devices attached to the vehicle frame or hull can direct this inbound flow rearward. The change in momentum thus generated will yield a propelling force. Calculations indicate that a 1-in.-wide pump opening at the rim of the wheels of a $\frac{1}{4}$-ton truck will yield the same flow area as the propeller on a DUKW. At this writing, the system has been designed and is presently under construction.

ARCHIMEDES SCREW

Though not a new idea (drawings from the fourth century show a ship propelled by an Archimedes screw), the concept of using a large, buoyant screw to propel a vehicle has recently received a great deal of attention in the form of the Marsh Screw Amphibian (Fig. 42). In semi-fluid conditions (marshes, mud flats, bayous), this vehicle has demonstrated its ability to propel itself where most other vehicles fail. It is therefore a quite suitable vehicle for operation in the transition zone between water and land.

FIGURE 42. Marsh Screw Amphibian (Courtesy Chrysler Corporation)

A basic principle behind the outstanding performance of this device, in environments where few other vehicles can operate, is represented by its ability to float a great deal of its weight in the propelling device. Ideally, the rotors should be large enough to float the entire vehicle, without the undercarriage touching water.

Comprehensive model-test programs were conducted at the University of Michigan and at Stevens Institute of Technology, to establish design criteria for operations both in water and on land.[34,35] In general, these tests established the existence of the following trends:

On Water:

(1) An optimum rotor configuration is not the same in all cases. The rotor must be matched to the vehicle and its propulsive system.

(2) In general, the 50-degree helix angle is better than the 30-degree and 40-degree angles tested.

(3) Performance improves with larger blade height.

(4) The performance of the two-lead screw is, overall, better than that of the one- or three-lead screw.

(5) Performance is optimum at a static trim angle of about +1 degree.*

(6) Optimum length/diameter ratio occurs near a value of 6.

(7) A purely cylindrical shape is definitely superior to a tapered one, especially at the stern.

(8) Progressive rearward increase of the helix angle has no beneficial effects.

(9) Tapering the front of the rotor hub has no effect.

(10) Cupped (as opposed to flat) blades are more efficient at high trim angles and high speeds.

*Positive static trim means bow up; negative, bow down.

(11) There is no basic difference between inboard and outboard turning of the two rotors on either side of the vehicle.

(12) Optimum rotor-spacing appears to be near 4 diameters.

(13) If shrouds are spaced so that they are close to the rotor, performance is improved.

(14) The model tests in water scale up well by Froude's Law.

Since the vehicle must be designed for amphibious operation, and since the propulsive elements are therefore common for both water and land operation, the following considerations must also apply:

In mud:[36]

(1) Optimum performance can be expected when the longitudinal CG is slightly aft of the rotor mid point.

(2) Maximum practical drawbar-pull occurs near 80-percent slip.

(3) The best helix angle (from among choices of 30, 40, and 50 degrees) is 30 degrees.

(4) Performance degrades with increasing blade height.

(5) Optimum length/diameter ratio is near 6.

(6) Model and full-size tests do not scale up well, because of the highly cohesive nature of the soil.

In sand:[36]

(1) Performance varies as described above for mud conditions, with but two exceptions, listed below as (2) and (3).

(2) Performance improves with increasing blade height.

(3) Model and full-size results correlate well.

The Marsh Screw Amphibian presents two serious problems which, as of this writing, have not yet been satisfactorily solved: it cannot operate well on hard surfaces such as roadways, and its performance on hard ice is erratic. To overcome these problems, several concepts are presently under study.

PADDLE WHEELS

Paddle wheels generate propulsive forces primarily through the generation of drag forces on paddle elements. Although propulsion efficiency as compared with that of other propulsion mechanisms is quite satisfactory, there are several disadvantages which have led to their disuse.

(1) Variable immersion under different loading conditions inhibits use on cargo vessels.

(2) The alternating rise of wheels above water level, while the ship is rolling, causes the ship to take an irregular course.

(3) There is a high risk of damage during rough weather.

(4) The low speed of the wheels requires large gear reductions from the high-speed machinery usually employed in modern ships.

(5) The installation of the wheels usually requires an increase in the ship's width, and this increases drag.

(6) High efficiency requires bulky wheels of larger diameter.

For certain applications, however, particularly for shallow-draft vessels operating in inland waterways (with drafts varying only slightly in service), the paddle wheel may still be practical. The ease of maneuvering a side-wheel paddle ship to a mooring is a decided advantage for operations in which maneuverability is important. In the case of amphibious military vehicles, operations are usually at a single, predetermined loading; and gear-reduction machinery already exists as part of the land-operation drive train. Thus paddle wheels may have a military use in inland water operations. This approach is presently being investigated by Stevens Institute under a recently initiated THEMIS program.

An analysis of paddle wheels for low-speed operation was developed recently by Gebers, Volpich, and Drappinger.[37,38,39,40] The basis of this analysis was a series of open-water model tests. Figure 43 shows the model of a feathered-float paddle wheel used in the tests. Note the linkages used in an attempt to keep the blades vertical while submerged.

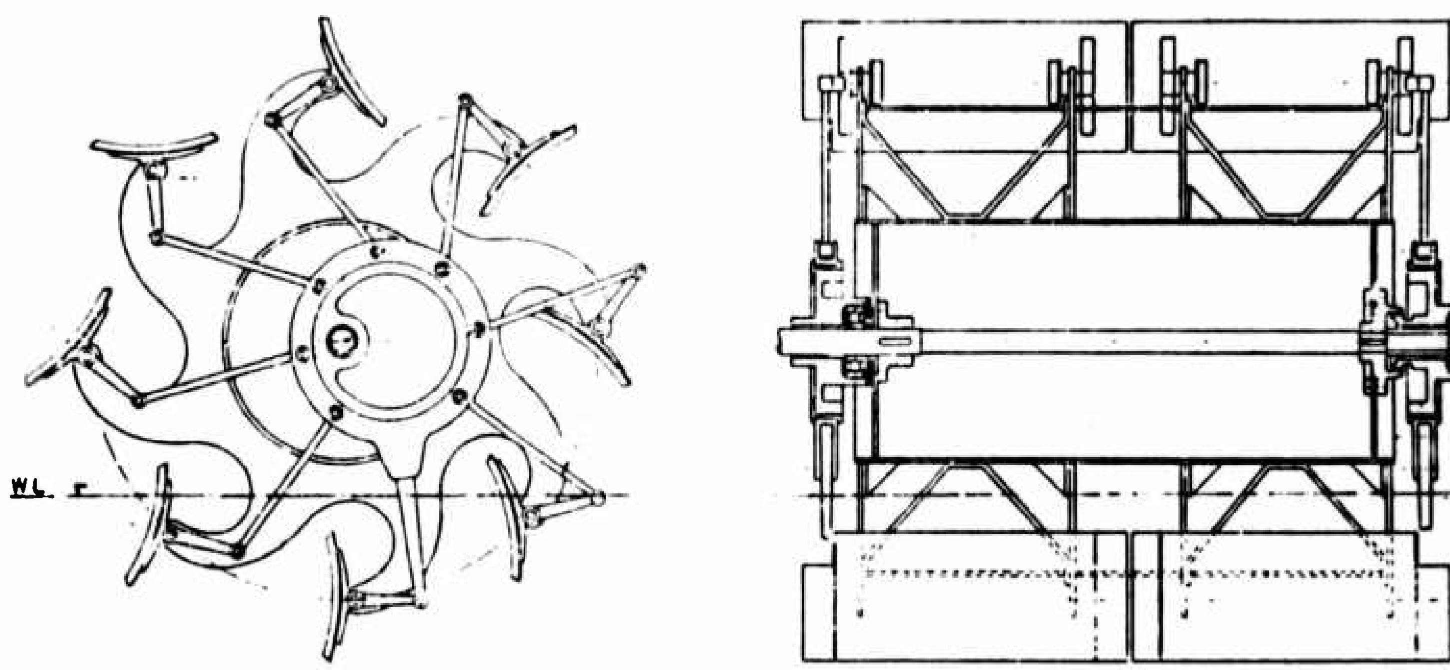

FIGURE 43. Model of a Feathered-Float Paddle Wheel

Certain of the conclusions derived from these tests are of general interest and are presented below.

(1) In general, the thrust and the required torque show an increase in proportion to rotational speed, up to a slip of approximately 35 percent. At this point a breakdown appears which is probably due to the losses that accompany the entrance and the exit of the floats (paddles), and to their mutual interference.

(2) The propulsion efficiency of a wheel with feathered floats can amount to approximately 80 percent; the efficiency of a wheel with fixed floats, to about 10 percent less.

(3) With equal rotational speed and equal forward speed, the thrust and the torque of a wheel are directly proportional to the width of the floats. Therefore, for a paddle wheel whose float width differs from that of known data, thrust and torque can be easily calculated.

(4) The thrust and torque of a wheel operating at two different immersions are in the same ratio to each other as the surfaces of the circular segments formed by the circumference of the wheel and the surface of the water. Thus, thrust and torque can be calculated for immersions of the wheel which differ from known test data.

The following equation may be used as a guide in estimating fixed radial-blade paddle-wheel performance for high-speed craft.

Let the following relationships exist (see Fig. 44):

D = outside diameter of wheel

b = wheel width

h = height of wheel axis above the free water surface

d = immersion, $D/2 - h$

r = effective radius to midpoint of blade, $(\frac{D}{2} + h)\frac{1}{2}$

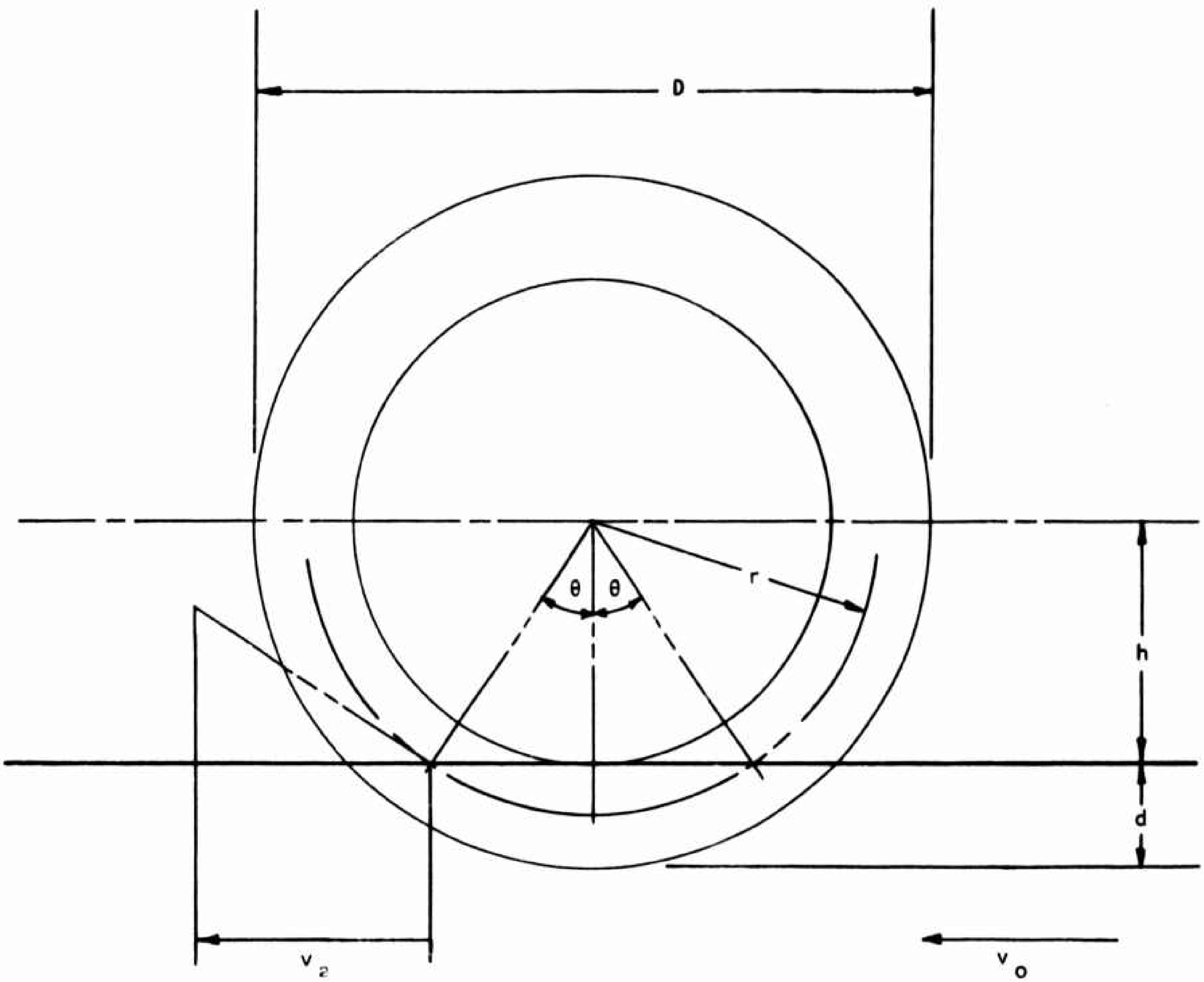

FIGURE 44. Geometry of Fixed Radial-Blade Paddle Wheel

θ = angle included by one-half the immersed arc at radius r,

$$\cos\theta = \frac{h}{r}$$

n = rotational speed of wheel, rps

v_o = inlet velocity of the water to the wheel

v_2 = exhaust velocity of the water away from the wheel

T = thrust

Q = torque

Then, from momentum theory, the thrust can be defined in terms of the inlet and exhaust velocities and the wheel geometry.

$$T = \left(\frac{v_o + v_2}{2}\right) bd\rho\,(v_2 - v_o) \tag{66}$$

$$= \frac{bd\rho}{2}\,(v_2{}^2 - v_o{}^2)$$

Rearranging,

$$v_2 = \left[\frac{2T}{bd\rho} + v_o{}^2\right]^{\frac{1}{2}} \tag{67}$$

If we assume that the relationship between rotational speed and exhaust velocity is

$$v_2 = 2\pi rn\cos\theta = 2\pi hn \tag{68}$$

then, transposing,

$$n = \frac{v_2}{2\pi h} \tag{69}$$

Torque may be conservatively approximated in terms of thrust, as

$$Q = \frac{Tr}{\cos\theta} = \frac{Tr^2}{h} \tag{70}$$

Solving for efficiency,

$$\eta_p = \frac{Tv_o}{2\pi nQ} = \frac{v_o}{v_2}\left(\frac{h}{r}\right)^2 \tag{71}$$

It will be noted that the efficiency is proportional to the ratio of the inlet and exhaust velocities and is very sensitive to the ratio of the height of the paddle axis to the effective radius.

Below is an example of what may be expected from a high-speed paddle-wheel-driven 12,000- to 15,000-lb amphibian.

Assume two stern wheels and a desired speed of 30 knots at which the drag of this vehicle (with a planing hull) is 2,000 lb. Also, assume that the wheel diameter (D) is 3.5 ft; that the width (b) is also 3.5 ft; and that the height of the wheel axis above the free water surface (h) is 1.25 feet.

$$r = \left(\frac{D}{2} + h\right)\frac{1}{2} = 1.5$$

$$d = \frac{D}{2} - h = 0.5$$

$$v_o = 30 \text{ knots} = 50.7 \text{ fps}$$

$$v_2 \left[\frac{2T}{bd\rho} + v_o^2\right]^{\frac{1}{2}} = \left[\frac{2(1000)}{(3.5)\ (0.5)\ (2)} + (50.7)^2\right]^{\frac{1}{2}} = 56.1 \text{ ft/sec}$$

$$n = \frac{v_2}{2\pi h} = \frac{56.1}{2\pi(1.25)} = 7.14 \text{ rps}; \quad n = (7.14)(60) = 428 \text{ rpm}$$

$$Q = \frac{Tr^2}{h} = \frac{(1000)\ (1.5)^2}{1.25} = 1800 \text{ ft-lb}$$

$$dhp = \frac{2\pi Qn}{550} = \frac{2\pi(1800)(7.14)}{550} = 147 \text{ hp/wheel}$$

$$ehp = \frac{Tv_o}{550} = \frac{(1000)(50.7)}{550} = 92$$

$$\eta_D = \frac{ehp}{dhp} = 0.63$$

Thus it should take 294 hp to drive this vehicle at 30 knots. This is clearly within present practical considerations.

TRACK PROPULSION

Tracks can be used as paddles for in-water propulsion, in lieu of or in addition to an auxiliary propeller or water jet (see Fig. 45). However, the complexity of the track system, which is incorporated for good ground performance, imparts many characteristics that degrade water performance. The hydrodynamic performance of vehicles based on this concept is generally poorer than that of vehicles for which other techniques are used. If, on the other hand, tracks are to be used anyway, for land operations, vehicles used primarily on land can profit from the use of the tracks for in-water propulsion.

The development of propulsion forces by such tracked systems depends on the same physical principle as that applied for other in-water thrusters -- the rearward acceleration of a mass of water. The efficiency with which this system operates therefore depends on the characteristics of the elements which are attached to the track to achieve this increase in momentum.

Tracks are designed to operate on land. The above noted complexity of track design affects track water performance most significantly where the location, width, height, spacing, and shape of the grouser are concerned. Some attempts have been made, recently, to capture the flow of water transported forward by the upper track sections, and to direct it rearward to increase efficiency. So far, the results of these efforts are inconclusive.

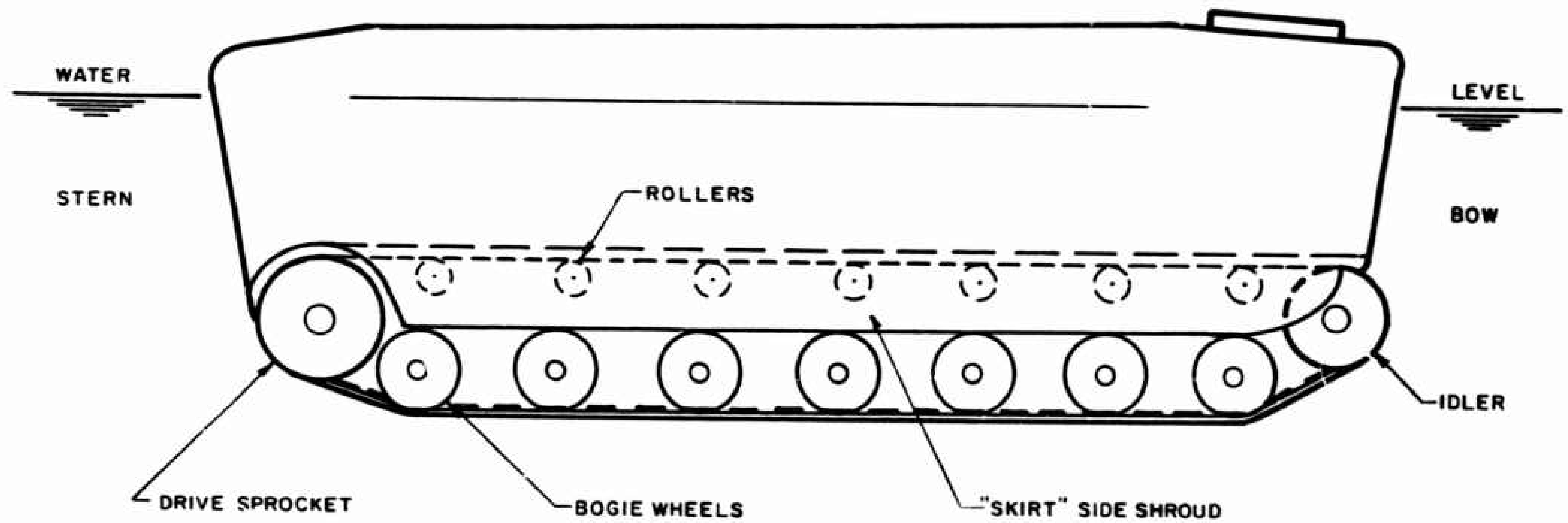

FIGURE 45. Sketch of Track Configuration (with some of components labeled) for Completely Submerged Track System

Quantitative data describing track performance are sparse. Model tests[41] indicate that the propulsive efficiency, η_D, of tracked vehicles is approximately 10 percent, compared to 60 percent and higher for propellers and paddle wheels. So there is much room for improvement. Studies on the drag characteristics of plates in tandem have been reported[42]; and an extensive investigation has been carried out on the effects of shrouds, return vanes, fenders, baffles, etc., mounted on the vehicle to control the flow near the tracks.[43] Both studies offer considerable design insight. Recently, the maximum water speed of the fully tracked armored personnel carrier M114 was increased 10 percent from 3.1 mph to 3.4 mph, by altering the design of front and rear fenders.[44] Several characteristics of importance are noted below, and their influence on performance is briefly described.

(1) Grouser location. Choosing the location of the thrust-producing element (referred to in the literature as the grouser, vane, cleat, paddle, or blade) on current amphibious vehicles, depends in large measure on avoiding the compromise of land performance. Among the locations on the track that have been investigated for suitability are bottom (the LVT-3), side (the LVTP-6), and center (the LVTP-5). Various experimental results are in conflict on the effect which grouser location has on hydrodynamic performance. Yet in no case did the propulsive efficiency exceed about 15 percent.

(2) Grouser width. The width of bottom grousers is generally limited to the track width, although effective width can of course be varied by the shape of the grouser. Center grousers, laid between the tracks, are also limited by track width. The width of side grousers, on the other hand, is primarily a function of other considerations -- vulnerability, structural loading on track, and design complexity.

(3) Grouser height. The allowable height of the grouser is limited by location considerations and on-land performance. No test data have been uncovered that indicated a preferred height. Since it is grouser area which is important in developing thrust, the combination of height and width is significant in design but is highly involved with shape, power transmission, and location characteristics.

(4) Grouser spacing. The spacing of grousers is a compromise involving sufficient separation to avoid

deleterious interactions between vanes and sufficient proximity to assure the generation of relatively steady propulsive force. Common practice has been to use spacings of about one-third the track width. However, this spacing appears to have no firm basis for support in either theory or experiment.[45]

(5) Grouser shape. Many and varied shapes have been proposed for grousers. These shapes range from simple flat rectangular elements to highly convoluted sections (primarily in bottom-grouser designs). Performance varies considerably across the range of designs, but no single configuration appears to offer such clear superiority as to make significant vehicle-performance improvements attainable.

(6) Submergence. Ideally, the submergence of the track should be such that the return path is clear of the water under fully loaded vehicle conditions. To assure the existence of this limiting submergence in the face of heavy wave activity, it is desirable (from the viewpoint of water performance) to design for approximately half of the track to be above water. This will tend to reduce the consideration of the vanes to wave-making resistance, and to remove the track from operation in regions of large turbulence. However, the incorporation of this feature into a vehicle in a practical manner is so difficult that the entire track is usually submerged.

(7) Flow pattern. Full-scale-data flow patterns are generally lacking or questionable; therefore, design must be based on empirical information. Without suitable information on the relationships between the physical characteristics of the track and the resultant flow pattern, it is possible only to point out the importance of avoiding areas of high turbulence and eddies and minimizing return flow in the track.

The usual procedures for studying the propulsion of screw-equipped vessels cannot be applied to tracked vehicles, since analyses of the former are based on resistance tests conducted with the propeller removed and on separate propeller propulsion tests in open water without a hull. The rationale for this technique is based on the assumption that there is little effect of one upon the other. Final tests are conducted to determine the propeller-hull interference effects, and the whole procedure provides considerable insight into where and how improvements can best be achieved (e.g., in the propeller, in the hull form, or in the location of the propeller on the hull). For track propulsion, however, the interference

between hull and tracks is much more extensive and complicated; hence open-water tests on tracks are almost useless. Removal of the tracks substantially alters the hull flow characteristics, so that tests involving removal tend to lose much of their significance.

Common procedure is to calculate the propulsive coefficient directly, from

$$\eta_D = \frac{R_T v_s}{2\pi Q n} \tag{38}$$

where the vehicle resistance R_T is determined with the tracks on the vehicle. Such a determination is not standard, however. Some investigators[43] measure the resistance R_T with the tracks moving, so that the velocity on the lower part of the track is equal to the forward speed v_s (also referred to as the zero slip condition). Others[44] measure vehicle resistance R_T with the tracks locked. Still others[41] use smoothed-out tracks of sheet metal, but attach wheels, sprockets, etc., to them. In the opinion of the authors, the zero-slip condition is most realistic.

Figure 46 is a conceptual drawing of a track with blades so cammed as to generate maximum thrust as they move downward and rearward, but minimum drag as they move forward. Such a system has been able to produce 20 lb of static thrust per input horsepower in a small model.[46] This is quite a creditable performance when compared with the curves presented in Figure 47.

Nuttall and Ehrlich[33] have recently concluded that designers are perhaps approaching the track-propulsion problem from the wrong direction. Most earlier efforts have been attempts to get a good rearward flow of water along the tracks, which would generate and improve forward thrust. Since it is better to accelerate a large quantity of water a little than to accelerate a small quantity of water a great deal (see Eq. [42]), Nuttall and Ehrlich have concluded that more benefit would derive if the water flow were directed inboard rather than rearward. This inboard flow could then be changed to a rearward flow, by static devices, thereby generating a forward thrust. This concept is presently under investigation (see discussion of wheel propulsion) and shows great promise of improvement for both wheel- and track-propulsion efficiency.

LEGEND:

A. Driven sprocket
B. Idler sprocket
C. Chain
D. Blade
E. Blade pin
F. Blade arm
G. Cam
H. Path
I. Roller

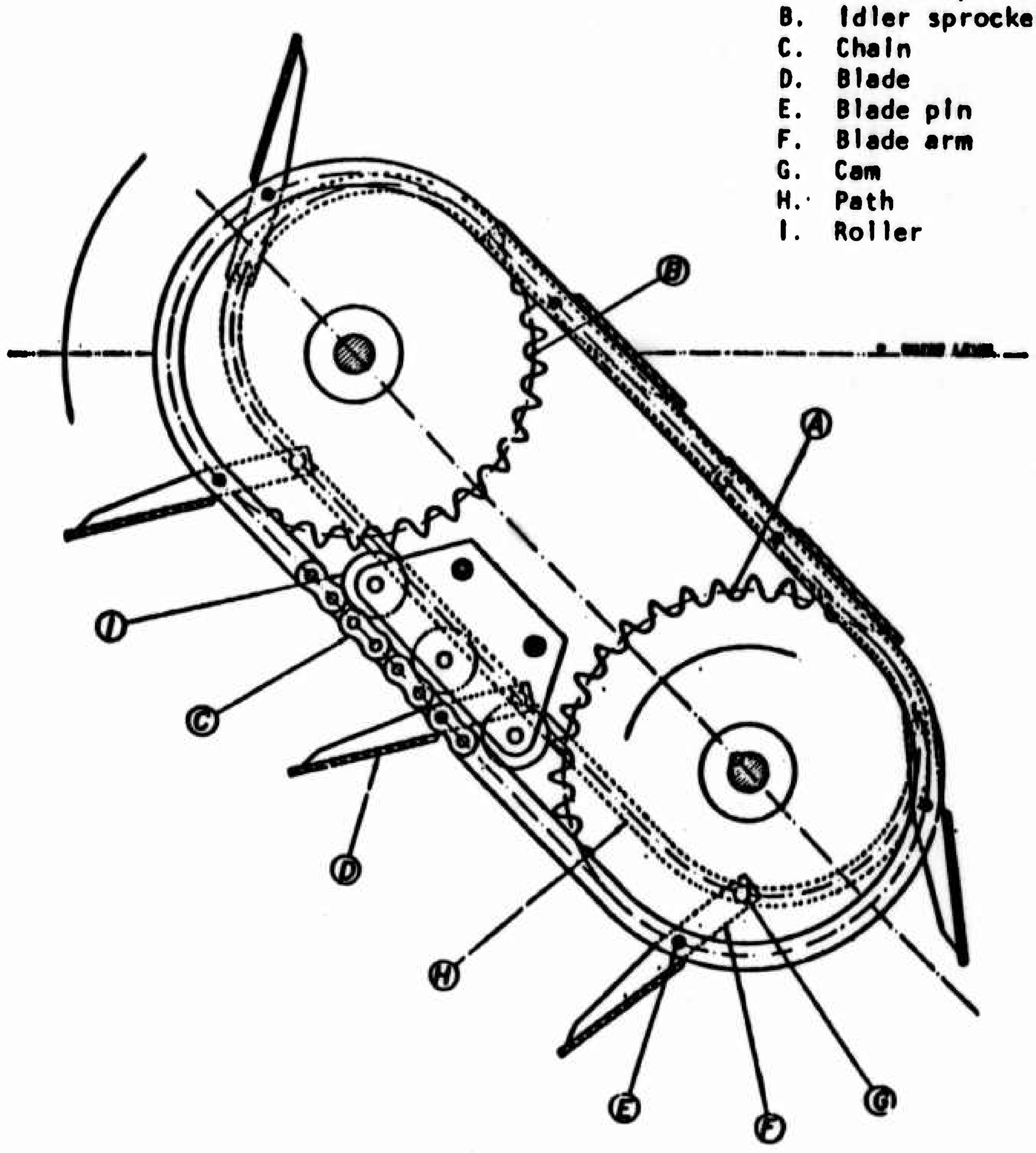

FIGURE 46. Schematic Drawing of the Paddle Track[46]

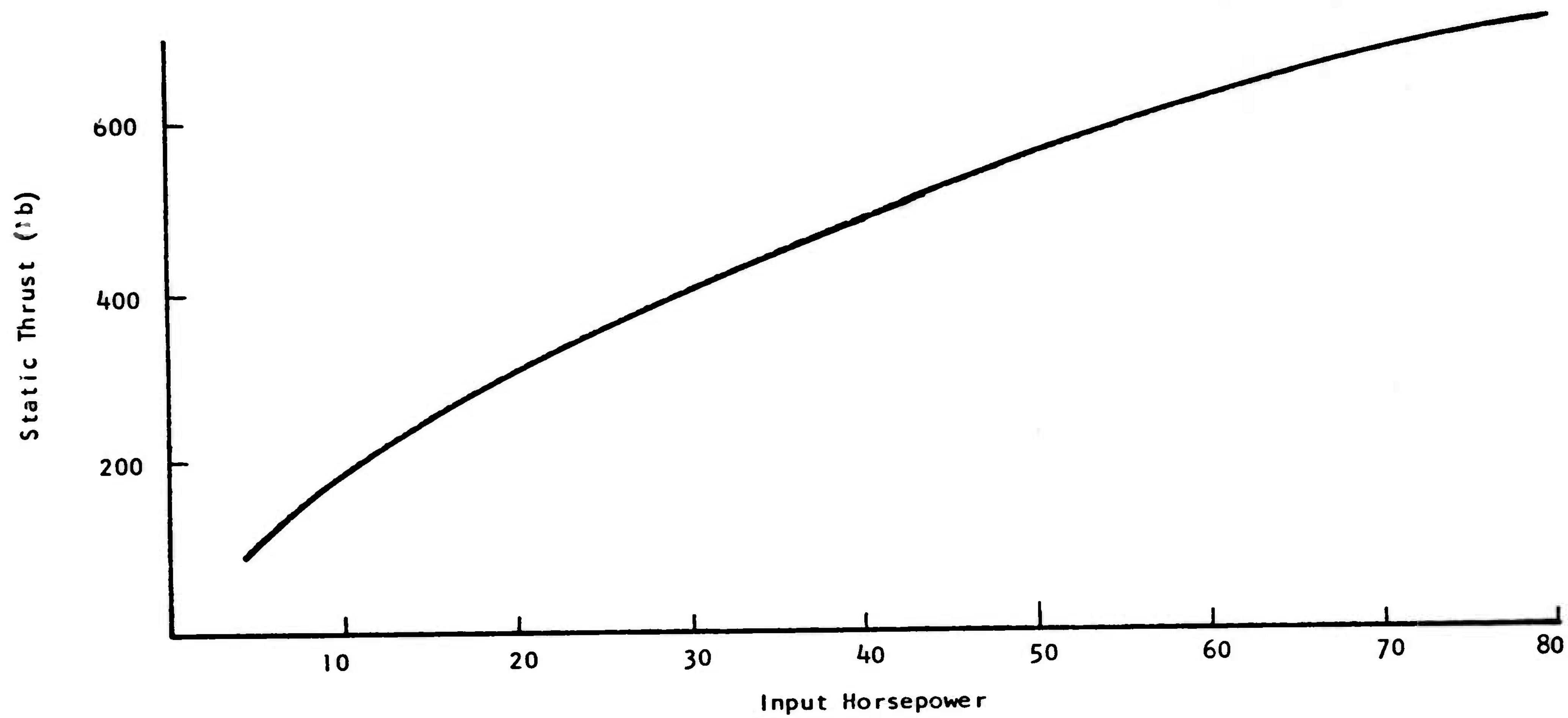

FIGURE 47. Typical Horsepower-Thrust Relationships for a Jet-Pump (Composite of the characteristics of a number of designs)

Recently a unique vehicle, termed the "Airoll" or "Marginal Terrain Vehicle" was developed. This vehicle uses air bags attached to a chain as a form of track (Fig. 48). These bags provide a large proportion of the floating vehicle's buoyancy and, in addition, act as paddle wheels for propulsion. Model tests[47] indicate that the propulsion effectiveness does not vary significantly with --

(1) Tire stiffness

(2) Tire diameter

(3) Tire spacing

(4) Shrouding

(5) Bag entrance angle

It does, however, vary with --

(1) Bag exit angle (lower angle gives better performance)

(2) Tire width/diameter ratio (performance improves with increasing aspect ratio)

(3) Vehicle trim (large bow-down static trim is advantageous)

Full-scale tests are still under study, but they indicate that the vehicle is capable of operating on a wide variety of terrains. Difficulties at this time appear to be associated with obstacle crossing, with exiting on slippery banks, and with providing comfortable riding conditions. Suggestions for the correction of these deficiencies have been put forward by Ehrlich, Nuttall, and Worden.[48]

ROWING

A little-considered but frequently used form of propulsion is man-powered rowing. Tests with 8-oared racing shells[49] indicate that each well-conditioned man of a well-trained crew delivers 0.269 hp of thrust over a sustained period of twenty minutes at 7 miles an hour. The standard engineer's handbook gives the power of an average man pulling an oar at near 0.1 horsepower. In terms of thrust, the trained oarsman is providing approximately 31 pounds; the average man, near 11.5 pounds.

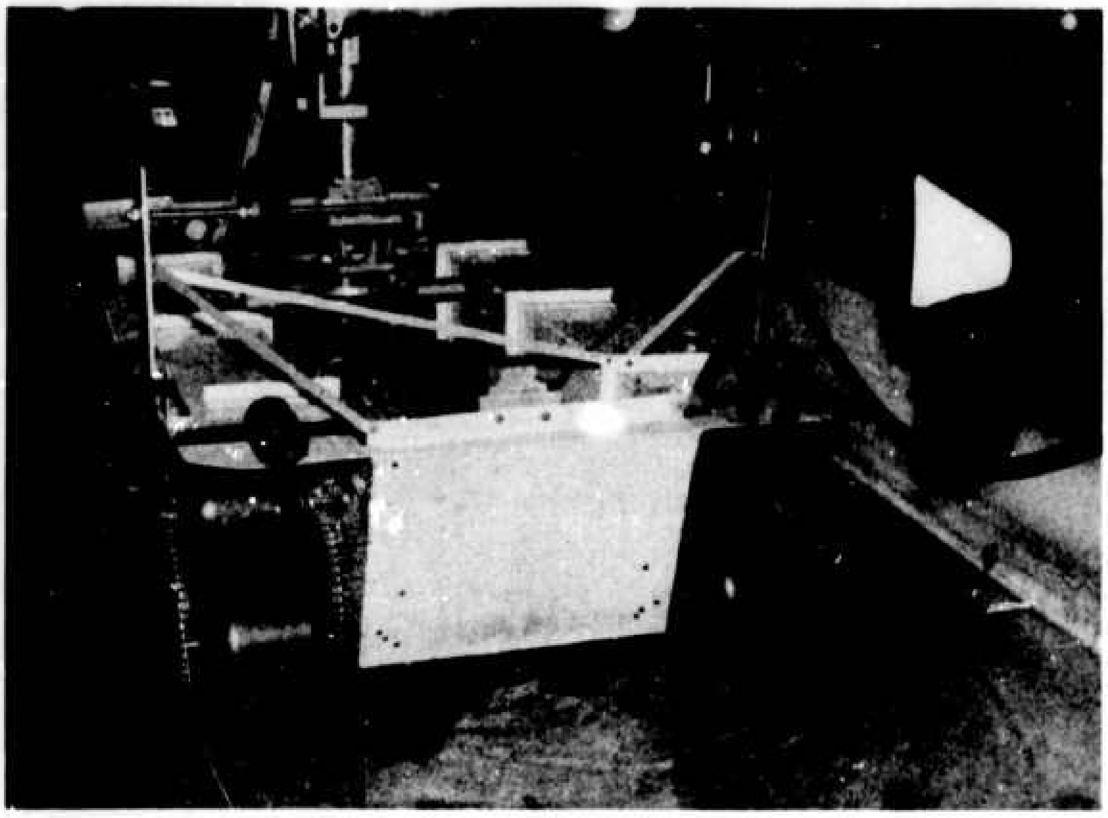

FIGURE 48a. $\frac{1}{4}$ Scale Model of the Air-Bag-Propelled XM759 Marginal Terrain Vehicle (MTV)

FIGURE 48b. Full-Scale Air-Bag-Propelled XM759 Marginal Terrain Vehicle (U. S. Army Photo)

Chapter 4
MANEUVERABILITY

CONTROLLABILITY

One of the most important properties which determine the maneuverability of an amphibious vehicle is controllability, or turning ease. Turning ease is the ability of the vehicle to change its direction of motion in accordance with the turning of the rudder or other device provided for the same purpose.

The direction of an amphibian's motion on water can, for example, be changed by any one or more of the following (not all-inclusive) methods:

(1) A rudder is the most common method. It is easy and inexpensive to install and to operate. In addition to controlling the vehicle, it contributes greatly to the vehicle's dynamic stability. When it is in its middle position, the water-stream flows symmetrically over both of its sides, and this tends to keep the vehicle traveling in a straight line. When the rudder is set at an inclination to the direction of motion, the water flow about the rudder is not symmetrical and the water pressures on the two sides are not equal. This difference in pressure produces a force on the rudder perpendicular to the surface of the blade, which tends to turn the vehicle in the same direction as the rudder. Rudders of large ships are usually foil-shaped, to reduce drag and give optimum performance. For amphibious vehicles, however, this is an unnecessary sophistication, and the use of a flat plate is recommended.

Since a boat in a steady turn pivots about a point near the bow, the rudder should always be located as far aft as possible, to obtain the largest turning lever arm.

Also, whenever possible, the rudder should be located in the flow race of the propelling track or other propulsion device. The higher flow rate in that region will often yield a turning force double that which would be developed in the general flow stream.

Experiments carried out by Gnomon[50] give the following useful equations for estimating the force normal to a rudder for ahead and astern conditions:

The <u>normal force coefficient</u> (C_N) is defined as

$$C_N = \frac{F}{\frac{\rho}{2} A v_s^2} \qquad (72)$$

where F = normal force (see below), lb

A = projected face area of rudder, sq ft

v_s = speed of vessel, fps

ρ = density of water, slugs/ft^3

δ = angle of rudder to centerline, deg

For vessels with the rudder at angle (δ) not in the propeller race,

$$F = 0.03\, A v_s^2\, \delta \qquad (73)$$

This equation applies explicitly at 35 degrees, and for angles less than 20 to 25 degrees the coefficient may rise to about 0.035.

The maximum normal force when the rudder is placed in the propeller race can be approximated by

$$F = 0.04\, A v_s^2\, \delta \qquad (74)$$

For vessels fitted with twin rudders behind wing propellers,

$$F = 0.041\, A v_s^2\, \delta \quad \text{(ahead)}$$

$$F = 0.037\, A v_s^2\, \delta \quad \text{(astern)}$$

(2) If the vehicle is equipped with a hydrojet, a turning moment can be generated by pivoting the exit nozzle. Some hydrojets are capable of pivoting far enough to make it possible for reverse thrust to be developed, so that the vehicle can back up without changing gears; in other cases, a similar effect is obtained by closing the rearward-facing exit nozzle and opening one facing forward.

(3) If the vehicle is equipped with two or more propulsors, it can be steered by proportioning the power input to the two devices or by restricting the flow from one of them. A common method of steering a vehicle with two hydrojets is to shut off the exit flow of one jet by a simple sliding door arrangement. Frequently, this same action opens a forward-facing exit port, to yield a rearward thrust on that side.

(4) Wheeled vehicles can be controlled simply by turning their steered wheels. The difference in drag created on the two sides will be sufficient to provide adequate steering for most circumstances, while the vehicle is under way. It will, however, be rather ineffective at low speeds.

(5) Multi-element (articulated) vehicles can be controlled by yawing one section with respect to the other. For a vehicle of three of more elements, yawing of the front section is more effective than yawing the rear.[11]

(6) Track vehicles can be steered in water just as they are steered on land -- by proportioning the speeds of the tracks on either side of the vehicle. Track-propelled vehicles are often especially clumsy while afloat. Since the tracks are so inefficient as propulsors, the turning moment generated by the differential track speed is usually quite small compared with the large hydrodynamic turning resistance and the very large yaw moment of inertia of the vehicle and its entrained water. Thus the vehicle, though maneuverable, has such a long response time that control frequently becomes quite difficult unless auxiliary devices are provided.

Vehicles which rely on tracks for a major portion of buoyancy (Airoll, Marginal Terrain Vehicle, PATA) frequently present steering difficulties, since tracks lose buoyancy with increasing speed. Thus a vehicle tends to list toward the faster-moving track. This list causes an unbalanced drag moment on the hull, which frequently is greater than the turning moment generated by the tracks. This causes the vehicle to turn in a direction <u>opposite</u> to that desired by the operator. Control vanes in the track race (water flow directly behind the track) and/or increased roll-stability may correct this deficiency.[47]

(7) Small reversible motors mounted transverse to the vehicle are frequently useful as control devices. They have the advantage, when actuated, of not detracting significantly from the propulsive operation of the vehicle (as do the previous systems mentioned), and they are effective when the vehicle has no forward motion.

DIRECTIONAL STABILITY

An amphibious vehicle with a low length/width ratio and a full bow contour is more subject to yawing than most boats. This is due to the forward location of the center of pressure and the short moment-arm of the rudder or other steering device. Vehicles with a forward trim and those with shallow draft are especially unsteady when under way. It is therefore advantageous to use a relatively large rudder for both directional stability and good control. For boats the size of amphibians, the area of the rudder should be near 1/30 the area of the projected underwater profile. If such a rudder size is not practical, the areas of two or more rudders may be added together to get the required size.

A rudder area greater than 1/30 the projected underwater profile will usually not increase maneuverability appreciably, but will tend to make the vehicle more directionally stable. A more stable vehicle will usually have a higher operational speed, since fewer rudder corrections, with the accompanying increase in drag, will be necessary.

The magnitude of the turning diameter is affected by (in order of importance) --

(1) The blade area of the rudder (a large area decreases the turning diameter).

(2) The angle at which the rudder is turned, up to a certain limit at which stall occurs (the turning circle decreases with increase in this angle).

(3) The shape of the underwater part of the hull and its dimensions (a wide hull will be less stable and easier to turn; a large draft will cause the vehicle to be more stable and harder to turn).

(4) The number and type of propelling devices and their location with respect to the rudder.

(5) Tunnel interference (a poorly designed rudder acting in a tunnel may restrict flow instead of generating turning forces).

(6) The rate of vehicle travel (the higher the speed, the larger the turning circle).

The calculation of the diameter of the turning circle is very complex, and is therefore usually determined by full-scale or model experiments. The turning diameters of most amphibious vehicles range from 25 to 75 feet. Ideally, the vehicle should complete its turning circle in three vehicle lengths, which is equivalent to approximately 9 deg/sec^2 turning acceleration at zero-degree heading and no angular velocity.

All amphibious vehicles should have some means of generating reverse thrust. They should also have means of generating a turning moment when the vehicle has no forward velocity. The former is essential for obvious reasons; the latter is essential to maintain the vehicle against a dock or shoreline, in a current.

Appendix
CHARACTERISTICS OF REPRESENTATIVE AMPHIBIANS, IN BRIEF

AMPHIBIOUS TANKS

	USA Mk1-LVT(A)1	USA Mk4-LVT(A)4	USA LVTN-6
Year issued	1943	1944	1955
Combat weight, tons	15.4	19	40
Crew, number	3-6	3-6	6
Over-all dimensions, ft			
Length	26	26	29
Width	10.8	10.8	11.7
Height	8.5	9.2	10.5
Maximum engine power, hp	250	250	810
Maximum speed on water, mph	7.5	5.9	7.5
Maximum negotiable grade, deg (exiting)	20	15	30
Maximum negotiable grade, deg (entering)	25	25	--
Rated cruising range on land, miles	200	150	120
Rated cruising range on water, miles	60	100	--
Unit ground pressure, psi	16.8	16.8	--
Type of drive on water	track	track	track

AMPHIBIOUS TRACKED ARMORED TROOP CARRIERS

	USA Mk2-LVT2	USA Mk2-LVT(A)2	USA Mk4-LVT4	USA Mk3-LVT3	USA M59	USA LVTP-5
Year issued	1942	1943	1944	--	1953	1954
Weight without load, tons	12.1	13.7	12.6	14.0	--	--
Payload capacity, including crew and landing force, tons	3.2	2.5	4.4	4.0	13.2	on land 9.0 on water 6.0
Combat weight, tons	15.3	16.2	15.9	18.0	21.0	35.0
Crew, persons	3-6	4	4-6	3	1	3
Landing force ferried, persons	--	24	24	24	11	23
Over-all dimensions, ft						
Length	21	21	21	24	18.5	29
Width	10.8	10.8	10.8	10.8	10.7	11.8
Height	8	8	8	8.5	7.3	8.8
Maximum engine power, hp	250	250	250	2 x 125	2 x 146	810
Maximum speed on water, mph	7.5	7.5	7.5	--	4.3	7
Maximum speed on land, mph	20	20	20	--	32	30
Maximum negotiable grade, deg	32	32	27	--	31	35
Range on land, miles	--	200	250	165	120	180
Range on water, miles	--	60	75	85	--	50
Unit ground pressure, psi	8.7	9.0	8.0	--	7.0	--
Type of drive on water	track	track	track	track	track	track

AMPHIBIOUS TRACKED PERSONNEL CARRIERS AND TRACTORS

	USA Mk-LVT1 "Alligator"	USA M29 "Weasel"	USA M76 "Otter"	GERMANY Tractor	FRANCE "Gabriel Voisin"
Weight in equipped condition, tons	15.9	3.0	4.4	16.5	3.1
Crew, persons	4	2-4	2-3	3	--
Cargo	24 persons	--	12 persons or 3000 lb	--	6 persons
Over-all dimensions, ft					
Length	21.5	10.5	--	29.5	--
Width	13.1	5.5	--	9.8	--
Height	8.1	5.9	--	10.3	--
Maximum engine power, hp	146	75	135	300	85
Maximum speed on water, mph	6.8	4	4	7.8	--
Maximum speed on land, mph	12	32	28	25	40
Maximum negotiable grade, deg	20	--	--	--	--
Range on land, miles	225	175	--	--	--
Unit ground pressure, psi	11.2	1.9	--	--	1.7
Type of drive in water	track	track	screw propeller	screw propeller	screw propeller

WHEELED AMPHIBIANS

	USA "Duck"	USA "Super Duck"	USA "Gull"	USA "Bark"	USA "Drake"	ENGLAND "Terrapin I"	ENGLAND "Terrapin II"
Weight in outfitted condition, tons	7.1	7.7	11	100	16.5	12.7	18.2
Payload capacity, tons	2.5	2.5	5	54 (or 203 men)	7-8	4.5	5
Over-all dimensions, ft							
Length	31	32	35	60	42	--	30.2
Width	8.2	8.2	10	26	9.8	--	8.8
Height	8.8	9.5	10	15.7	11	--	6.6
Maximum speed on land, mph	50	50	50	10	45	20	25
Maximum speed on water, mph	5.9	6.8	9	6.5	9	5.6	6.2
Maximum negotiable grade, deg	30	31	31	$26\frac{1}{2}^{\circ}$	--	--	--
Engine maximum power, hp	90	145	--	4 x 165	2 x 155	2 x 85	--
Type of drive in water	screw propeller	screw propeller	screw propeller	screw propellers	screw propellers	screw propellers	screw propeller
Propeller diameter, in.	$18\frac{1}{2}$	--	--	--	--	$18\frac{1}{2}$	27

REFERENCES

1. U. S. Army Contract No. DA-30-069-AMC-789(T).

2. DL Proposal 408, June 1965.

3. ROSSELL, H. E. and CHAPMAN, L. B., Principles of Naval Architecture, Vol. II, Society of Naval Architects and Marine Engineers (SNAME), New York, 1939.

4. LAMB, H., Hydrodynamics, Dover Publications, Inc., New York, 1945.

5. BEKKER, M. G., Theory of Land Locomotion, University of Michigan Press, Ann Arbor, Mich., 1960.

6. Staff, Land Locomotion Laboratory, "Research Report 5," U. S. Army Tank-Automotive Command (ATAC), 1958.

7. "Trafficability of Soils; A Summary of Trafficability Studies Through 1955," Waterways Experiment Station TM3-331, 1957.

8. Staff, Land Locomotion Laboratory, "Research Report 6," Technical Report 9560, ATAC, 1966.

9. United States Coast Guard Rules and Regulations for Passenger Vessels, Subchapter H, CG-256, May 1966.

10. FULLER, S. H., letter dated 10 March 1964 from HQ, U. S. Army Tank-Automotive Center, SMOTA-RR to CG, U. S. Army Materiel Command, AMCRD-DM-G.

11. DUGOFF, H., EHRLICH, I. R., KAMM, I. O., NUTTALL, C. J., Jr., and SCHWARTZ, R. B., "Coupled Mobility Devices," Davidson Laboratory Report 1164, Stevens Institute of Technology (SIT), 1964.

12. HAY, D. A. and RUNYON, J. P., "Photographs and Resistance Measurements of Semi-Submersed Right Parallelepipedons," Princeton University Job Order 4, Contract Nobs 34006, 1947.

13. WORDEN, G., "Wavelet Swamping Study; A Method for Predicting Rate of Shipping Water," Davidson Laboratory Report 1254, SIT, November 1967.

14. ROBERTSON, J. M., Hydrodynamics in Theory and Application, Prentice-Hall Inc., Englewood Cliffs, N. J., 1965.

15. Special tests, Project 439, Davidson Laboratory, SIT, May 1964.

16. GONDERT, T. R. and FINELLI, J. P., "Towing Tests of a Proposed Lighter, 5-Ton, Amphibious," Davidson Laboratory LR-701, SIT, June 1958.

17. KAMM, I. O., "What Price Water Speed Improvements of Amphibians," Proc., Second International Conference of International Society for Terrain Vehicle Systems, 1966 (Davidson Laboratory Note 749, SIT).

18. KAMM, I. O. and UYGUR, D. M., "Towing Tests of a 1/12-Scale Amphibious Sea-Train Concept," Davidson Laboratory LR-724, SIT, May 1966.

19. VAN DYCK, R. L. and EHRLICH, I. R., "Drag Studies of Coupled Amphibians," Davidson Laboratory R-1137, SIT, July 1966.

20. KAMM, I. O. and UYGUR, D. M., "Smooth- and Rough-Water Tests of a 1/10-Scale Model High-Chine V-Bottom Planing Hull," Davidson Laboratory LR-1146, SIT, May 1966.

21. GONDERT, T. R. and FINELLI, J. P., "Towing Tests of Two Scale Models of the Amphibious Drake (XM-157)" Davidson Laboratory LR-713, SIT, January 1957.

22. CLEMENT, E. P. and BLOUNT, D. L., "Resistance Tests of a Systematic Series of Planing-Hull Forms," Trans. SNAME, Vol. 71, 1963.

23. SAVITSKY, D., "On the Seakeeping of Planing Hulls," SNAME Presentation, May 1966 (Davidson Laboratory Note 739, SIT).

24. KAMM, I. O. and FINELLI, J. P., "Towing Tests of a 1/10-Scale Model DUKW Equipped with Hydrofoils," Davidson Laboratory LR-636, SIT, February 1957.

25. KAMM, I. O. and FINELLI, J. P., "Resistance and Trim Characteristics of Two Hydrofoil-Equipped Hulls with the Same Over-all Dimensions as the DUKW," Davidson Laboratory LR-695, SIT, April 1958.

26. HAY, D. A., "Effect of Varying One End of Barge Forms with Simple Geometrically Shaped Ends," Princeton University Job Order 3, Contract Nobs 34006, 1946.

27. HOERNER, S. F., Fluid Dynamic Drag (published by the author), Midland Park, N. J., 1958.

28. GONDERT, T. R., "Towing, Water Entry and Static-Stability Tests of the $\frac{1}{4}$-Ton 4x4 Truck," Davidson Laboratory LR-866, SIT, August 1961.

29. VAN MANEN, J. D., "Fundamentals of Ship Resistance and Propulsion; Part B, Propulsion," Publication 132a, Netherlands Ship Model Basin (reprint from International Ship Building Progress), 1957.

30. HENDERSON, R. E., McMAHON, J. F., and WISLICENUS, G. F., "A Method for the Design of Pump Jets," Pennsylvania State University, May 1964.

31. RYMISZEWSKI, A. J., "Improving the Water Speed of Wheeled Vehicles," Journal of Terramechanics, Vol. 1, No. 1, 1964.

32. DUGOFF, H., "Studies in Totally Submerged Wheeled Propulsion," unpublished.

33. Davidson Laboratory Proposal P-556, "Development of Wheel Disc Pump for the Propulsion of Wheeled Floating Vehicles," 1967.

34. KRESS, R. F. and NEUMEYER, M. J., "Design Manual for Buoyant Screw Propulsion," Chrysler Corporation, Defense Operations Division, 1965.

35. DUGOFF, H. and EHRLICH, I. R., "Systematic Tests of Marsh-Screw-Rotor Models in Soils," Davidson Laboratory, SIT, Report 867, 1965.

36. DUGOFF, H. and EHRLICH, I. R., "Model Tests of Buoyant Screw-Rotor Configuration," Proc., Second International Conference of International Society for Terrain-Vehicle Systems (Davidson Laboratory Note 746), 1966.

37. GEBERS, F., "Das Schaufelrad in Modelsversuch," Springer-Verlag, Wien, 1952.

38. KRAPPINGER, O., "Schaufelradberechnung," Schifftechnic, 1954.

39. VOLPICH, H., "Paddle Wheels, Part I," Institute of Engineering and Shipbuilding, Scotland, 1955.

40. VOLPICH, H. and BRIDGE, I. C., "Paddle Wheels; Part II, Systematic Model Experiments," Institute of Engineering and Shipbuilding, Scotland (paper), 13 March 1956.

41. EHRLICH, I. R. and VAN DYCK, R. L., "Drag Studies of Coupled Amphibians," Davidson Laboratory Report 1137, SIT, July 1966.

42. EHRLICH, I. R., MERCIER, J. A., and TANAKA, I., "Studies in Hydrodynamic Track Propulsion," Davidson Laboratory Report 1136, SIT, September 1966.

43. MOSS, J. L. and SLATER, D. A., "EHP and DHP Model Test Results on LVTPCII Amphibious Vehicle for Ingersoll Kalamazoo Division, Borg-Warner Corp.," University of Michigan Ship Hydrodynamics Laboratory, November 1962.

44. HECKER, L. G. and NUTTALL, C. J., Jr., "Summary Report Covering (1) Track Propelled Amphibians and Conversions That Sparkman & Stephens, Inc. Has Assisted in Developing and (2) Use of Completely Submerged Tracks for Propulsion in Water," Sparkman & Stephens, Inc. Project 500, May 1944 (CONFIDENTIAL).

45. LAZAREK, W. A., "Report of USATECOM Project 1D-3251-07, Product Improvement Test of Water Propulsion Aids for Carrier, Personnel, Full-Tracked, Armored, T114," Report DPS-1083, U. S. Army Test and Evaluation Command, Aberdeen Proving Ground, Maryland, September 1963.

46. PAVLICS, F., "A New Booster of Lift and Thrust for Amphibian Vehicles," Report 32, Land Locomotion Laboratory, Detroit Arsenal, 1958.

47. DUGOFF, H., EHRLICH, I. R., and SCHAEFER, R., "Interim Report on the Water Performance of 1½-Ton Marginal Terrain Vehicle," Davidson Laboratory Note N-763, SIT, 1967.

48. EHRLICH, I. R., NUTTALL, C. J., and WORDEN, G., "Concept Feasibility Study of an Articulated Air-Roll Vehicle (Summary Progress Report)," Davidson Laboratory Report 1283, SIT, August 1968.

49. RODGERS, ADM., "Notes on the Design of and Propulsion of the 8-Oared Shell," USEMB Report 117, August 1925.

50. GAWN, R. W. L., "Steering Experiments, Part I," Royal Institute of Naval Architecture, 1943.

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13. ABSTRACT

As the first of the three volumes of the publication entitled "Studies of Off-Road Vehicles in a Riverine Environment," this report attempts to assemble the vehicle-environment relationships which have been developed and established in other areas of research and which are applicable to swimming-vehicle design. Many generalizations and approximations have been employed, so that the work will be of practical interest to the amphibious-vehicle designer.

The study neglects relationships between the vehicle and air, since air is relatively insignificant at the general range of speeds under consideration.

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